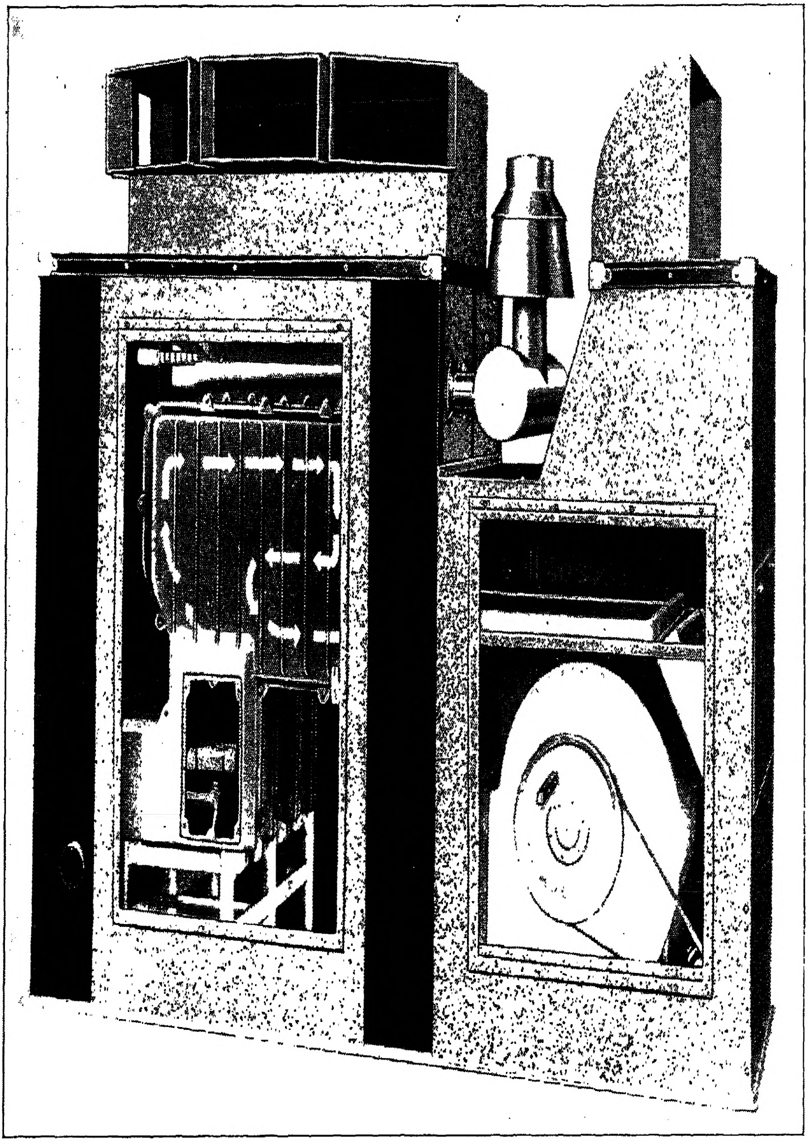

GAS-FIRED WINTER AIR-CONDITIONING FURNACE

Courtesy of The Henry Furnace & Foundry Company, Cleveland, Ohio

HEATING AND VENTILATING

AIR CONDITIONING

A Home-Study Course and General Reference Work on the Principles, Design, Selection, and Application of Heating and Air-Conditioning Appliances and Systems for Residential, Commercial, Industrial Use.

Illustrated

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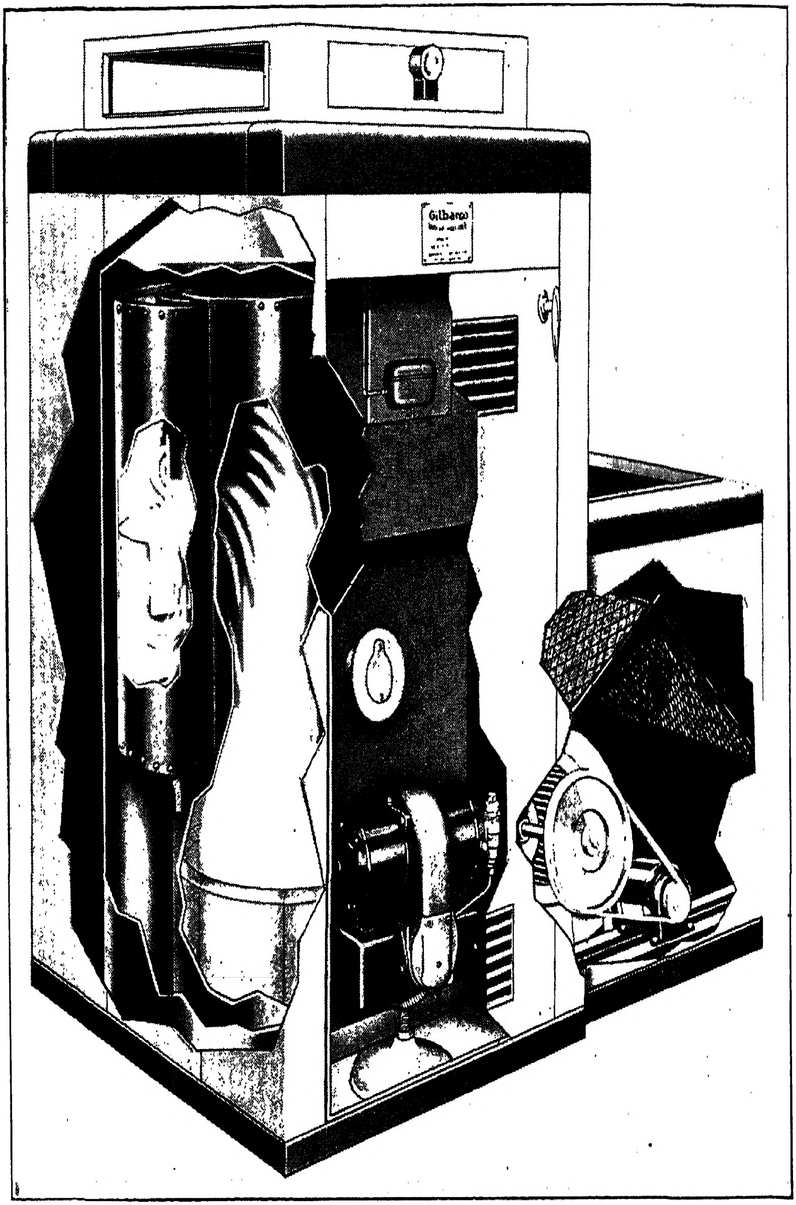

TYPICAL AIR CONDITIONER
Courtesy of Gilbert & Barker Mfg. Co.

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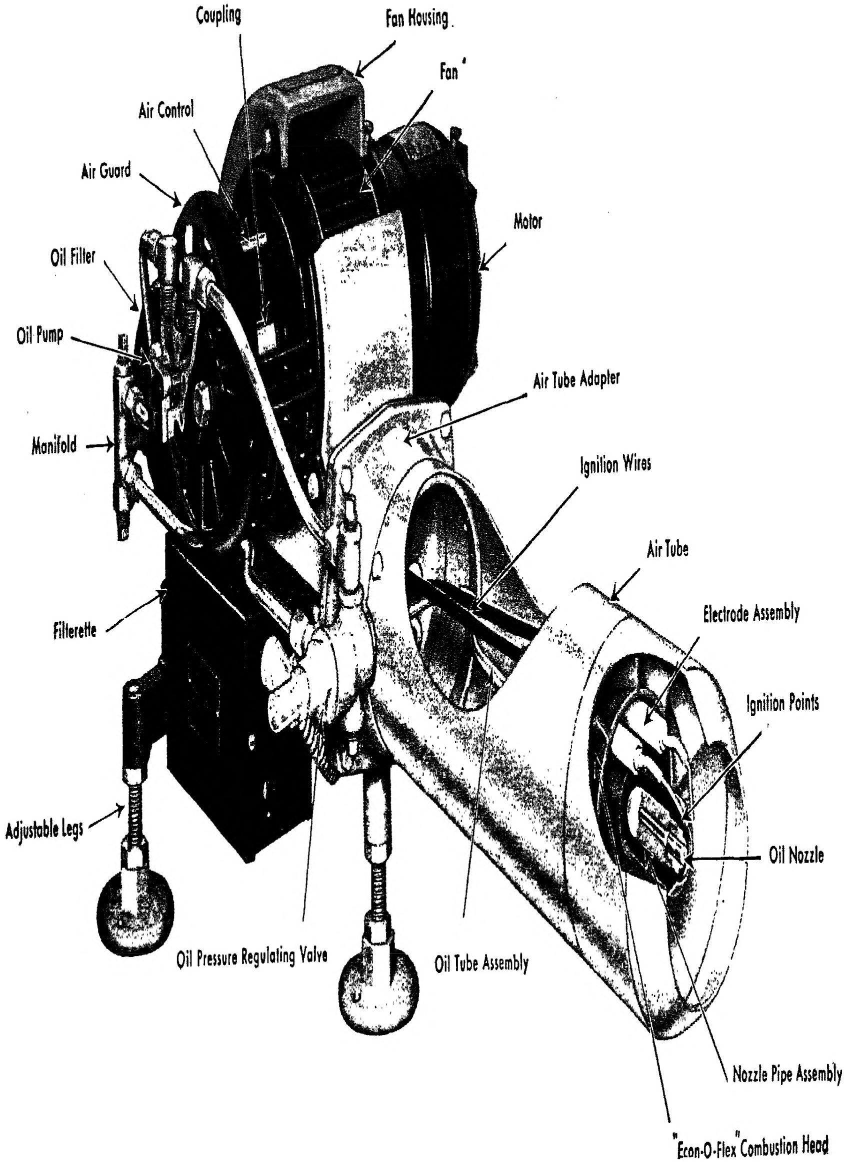

OIL BURNER

Courtesy of Gilbert & Barker Mfg. Co., Springfield, Mass.

FOREWORD

★AIR CONDITIONING is not a new idea. As early as 1911 Willis H. Carrier had formulated some of the principles and laws which are being used in present day air-conditioning engineering. However, he probably did not dream of the possibilities in the field to which he contributed.

★At first, air conditioning was developed only for use in factories where the control of humidity and where summer cooling permitted the continuation of processes previously confined to the cool months of the year. Gradually, the new applications of old principles have brought the possibility of year-round ideal manufacturing conditions in industrial plants.

★The success achieved in industrial air conditioning suggested the possibilities in conditioning primarily for comfort. Theatres, restaurants, and stores—which always had suffered a hot-weather decline in business—offered a fertile field. In theatres, the cooled air increased business during the summer to a point never before known. Restaurants and stores also enjoyed the new summer prosperity. Thus air conditioning for comfort was established. Public demand for greater summer comfort, and the success of air-conditioning engineers in achieving it, gave impetus to the work of residential air conditioning.

★Few industries have enjoyed as rapid a development as air conditioning, and few industries have such potentialities. Air conditioning includes the treatment of air in one or a combination of several of the following ways: heating, cooling, humidifying, dehumidifying, ventilating, and cleaning. The air-conditioning industry is demanding trained men to carry on the work of improving and installing all types of systems, from those used in factories to those used in homes. The training necessary for this work is peculiar to the industry and highly technical.

★These volumes aim to meet the needs of men who, with the demand of the industry for specialized training in mind, look forward to joining those who are building the air-conditioning industry.

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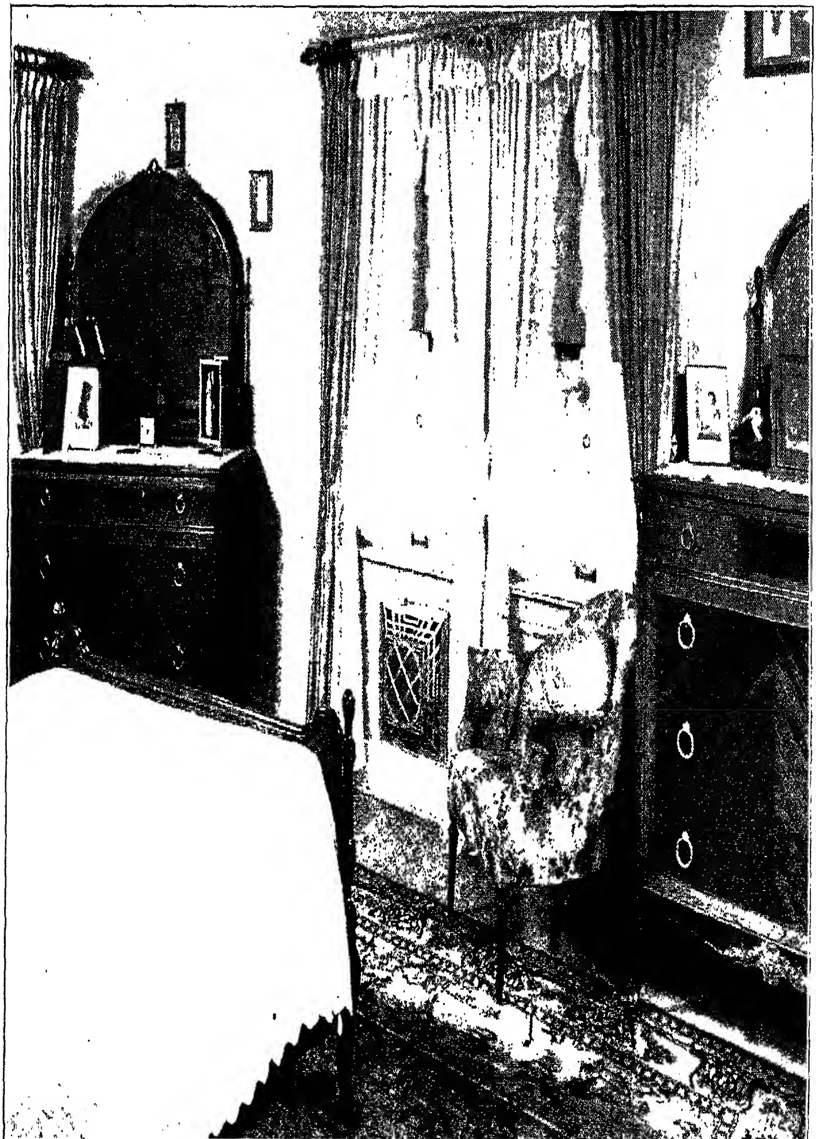
*For page numbers of this volume, see top of pages.

†For professional standing of authors, see list of Authors and Collaborators at front of volume.



SHOWING ELECTRIC HEATER UNIT INSTALLED IN A RESIDENCE

Courtesy of Electric Heater Company, Mishawaka, Indiana

FURNACES AND UNIT HEATERS

CHAPTER I

INTRODUCTION

The man who discovered fire remains an unsung hero, as he evidently existed long before the times recorded by history. Before its discovery man's movements on the face of this planet were more or less circumscribed by temperature and food supply. With the discovery of fire, however, man became less of a "lone wolf." Fire with its mystery, its warmth, and its protection against animals, helped to divide men into groups each having the germ of a clan organization.

When the means of providing fire at will were discovered, fire preservation sank into a lower place, and it became possible for man to migrate into different zones. The development of the uses of fire and its twin sister—steam—added much to the enrichment of life and aided men to settle in the waste places of the earth and make them habitable. Modern civilization seems to have grown on the ability of man, by the aid of mechanical means, to adjust himself to the varying temperatures which the earth forces upon him. Keeping warm has been a prime necessity for the preservation of life, therefore there have always been thousands of minds engaged in the problems of improving and inventing appliances for the more effective use of fire in helping to keep us warm.

Heating of occupied interior spaces has progressed from the rude open fire, dating back before written history, through many centuries of changes and improvements. These include stoves, hard-coal burners, gravity furnaces, mechanical furnaces, and finally this new science of air conditioning. Each new stage in this development of heating apparatus brought about added comfort, economy, and convenience. In this modern age it is possible to heat and air-condition entire buildings during a full heating season with no more effort than that required to light a small pilot light at the beginning of the heating season; or, the paradox of the situation, it is possible

to cool the same building during the summer months with some of the same apparatus, using heat exchange as the primary principle, and with little more effort than is required to push an electric switch.

Thus the poorly heated homes of our great-grandfathers are being replaced by modern insulated dwellings where the air is cleaned and where heating and humidifying or cooling and dehumidifying are all automatically accomplished.

The modern home may have the benefits of healthful, comfortable, clean, and invigorating air without any of the dirt of fuel and ash handling, and without poor heat distribution, dry air, and generally unsatisfactory conditions that prevailed before the advent of the air-conditioning furnace.

The stoves or hard-coal burners were the simplest of the first heating devices and relied upon the heat being diffused by radiation and convection directly to the objects and air in the room. It was almost impossible to maintain a steady temperature because of frequent firing, and the distribution of heat, in terms of intensity, was proportional to the distance away from the stove. The handling of fuel and ashes caused considerable dirt and required near-by storage of fuel and frequent attention.

The first furnaces introduced a distinct advantage, in that an entire building could be heated from one central source which was conveniently located in the basement. These furnaces required less care, eliminated the dirt and litter of stoves in living rooms, provided much more even temperatures, gave better distribution, supplied considerable ventilation and, in other words, provided more healthful living conditions. There were, however, many points of disadvantage due mainly to lack of actual knowledge regarding heat control and the effects of humidity. Certain rooms could not be heated so well as others, gas (product of combustion) leaked through the fire boxes and found its way into the air supply, and the air delivered to the rooms was so dry as to cause discomfort to people and to do much damage to furniture.

The later furnaces, still employing the gravity principle, had the benefits of considerable knowledge and experience in their construction and installation. An important feature for securing humidification was also added. The first of these later furnaces were better designed, their efficiencies were much higher and the gases could

be controlled. Considerable additional knowledge had been gained relative to proper size of supply pipes and their location. The newer principle of cold-air return pipes and their locations was also introduced. Recirculation of air and proper ventilation were developed and actual air changes were calculated in connection with the heating requirements, which further improved the situation. Some humidification was made possible by placing an open tank of water in the furnace casing where the warm-air currents could absorb some of the moisture very much needed in all the heated areas.

Still later types of gravity furnaces contained further structural improvements which reduced their bulk and at the same time raised their heating efficiency. Designs of basement supply pipes and stacks were improved and it became much easier to heat *all* rooms evenly. The selection of furnace sizes has been made more accurate, doing away with the chances of having furnaces too small or too large. Humidification was improved by new designs in humidifier tanks and by changes in their locations, with the result that the air was humidified to a point much nearer the desired degree. Automatic firing devices and semiautomatic controls made their appearance in definitely noticeable numbers. Gas and oil burners passed the experimental stage and coal stokers automatically controlled by thermostats came into common use.

The present-day gravity furnace is introducing many more refinements in the items already mentioned, in addition to the filtration of air and mechanical humidification. Full automatic control of firing, temperature, humidity and filtered air is now common. The present-day furnace owner may light a pilot light late in the fall and practically forget about the furnace for the duration of the heating season. He is assured of ample warmth, automatic control of day and night temperatures, proper humidity, and clean healthy air.

By degrees it became known to engineers and manufacturers that greater comforts, controls, filtration, humidity, and even summer cooling could be accomplished by forcing the circulation of air instead of depending upon gravity for all circulation. The first experiments in this line consisted of installing a fan or blower in one of the large cold-air return pipes at a place near the point where the pipe entered the furnace casing. The success of trials, while meeting with many obstacles, nevertheless indicated the right direction

for further experiments. These have led up to the automatic and positive mechanical furnace of today, where the amount of air for every room is calculated and supplied in exact and required amounts, where circulation is according to the need of each room, where humidity requirements are 100 per cent fulfilled, and where filters remove even the smallest amounts of dirt and other impurities. Automatic control is 100 per cent and summer cooling is possible using either natural air, refrigeration, ice, or cold water. Dehumidification now makes possible not only the cooling of summer air, but also the removal of the excess moisture which makes for uncomfortable conditions.

True air conditioning for both winter and summer is now possible by using mechanical furnaces.

As improvements in furnaces brought more and more comforts and conveniences, their selection and installation principles became more complicated until at present only a qualified heating engineer can successfully select and install them. This fact has led to an acknowledged need for special education among furnace dealers and others interested in furnaces, and it is to that end that this book is dedicated. Approved methods and typical apparatus are employed and illustrated throughout the following chapters, together with the type of examples that illustrate the common problems encountered continually by the heating engineer.

In addition to furnaces, other methods of supplying warm air have been devised which, while not as efficient in an all-around manner as furnaces, supply an occasional or limited need.

Electric Heating. While not new, electric heating has lately been developed in conjunction with cheaper electricity, until it can be used economically for occasional or even major heating. The small space required by the heaters, the positive supply of heat and circulation, and the lack of fuel responsibilities have caused electric heating to be accepted generally. Electric heating compares favorably with other means of heating, and proves as economical where energy rates do not exceed 3 cents a kilowatt.

Unit Heating. Unit heating has become popular because small stores, restaurants, industrial plants, etc., can be amply heated by this means without the expense of installing a central heating system. Units are manufactured on a production basis and therefore can be

produced much cheaper than a central system for the same structure. In addition, the installation charges are much less because the units are shipped complete, ready for hanging and operation. Unit heating may be accomplished by steam, hot water, gas, or electricity. The steam and hot-water types require steam or hot water, which can be supplied by small boilers remotely located. The heating advantages of units are that the heated air originates at the ceiling level, or slightly below it, and a system of louvres insures even distribution at the floor and breathing levels without spots of excess heat, such as caused by an ordinary radiator or register. Units in constant use require a minimum of attention, have high strength with but little weight, and are capable of handling large volumes of air which is discharged at moderate temperatures.

Mechanical Furnace Systems. Mechanical furnace systems have an advantage over gravity or old style furnaces: forced circulation. Their use solves problems of rooms "hard to heat," allows the furnace to be placed anywhere in the basement, and reduces duct or leader sizes materially. Thus the basement in a residence may be planned for many uses without the objection of a centrally located furnace with its large round leaders and cold-air pipes taking up a great amount of room. The mechanical system is well-adapted to automatic control.

Air-Conditioning Furnaces. Air-conditioning furnaces are an improvement over the mechanical furnace because in addition to forcing the air to all interior spaces, the air is cleaned, washed, and humidified in winter, and cleaned, cooled, and dehumidified in summer. There are many intermediate furnaces of types in which filtering, washing, humidifying, dehumidifying, cooling, etc., are done in varying combinations. Such furnaces are easily adapted to either coal, gas, or oil fuel and automatic control.

Split Systems. Some larger buildings, such as office buildings, are heated by steam radiators while a fan system ventilates by supplying air at about room temperatures—air which has been cleaned, washed, humidified, etc. In such a system the radiators keep the heat up to the required temperatures, or the system may be modified so that the radiators supply only a portion of the required heat and the fan system supplies the remaining heat plus ventilation.

Air-Conditioning Systems. In these systems all heating and

ventilating is supplied by means of fans, and the air is heated, circulated, washed, humidified, dehumidified, cooled, etc., as the season demands. Such systems employ a boiler for supplying steam to heating coils but are much different from air-conditioning furnaces. Air-conditioning systems are generally used where full all-year conditioning is required, although air-conditioning furnaces can be made to fill the same demands.

Unit Air Conditioners. Unit air conditioners are designed to provide heat, cooling, ventilation, or any combination of these items, for individual rooms or other comparatively small enclosures. There are window types, floor types, ceiling types, and stationary or movable types. The unit conditioners are especially suitable for buildings which were built before the era of air conditioning. Multiple units can be operated from one compressor or one steam or hot-water supply.

CHAPTER II

PRINCIPLES OF VENTILATION

Closely connected with the subject of heating is the problem of maintaining air of a certain standard of purity in the various buildings occupied.

The introduction of pure air can be done properly only in connection with some system of heating; and no system of heating is complete without a supply of pure air, depending in amount upon the kind of building and the purpose for which it is used.

Composition of the Atmosphere. Atmospheric air is not a simple substance but a mechanical mixture. Oxygen and nitrogen, the principal constituents, are present in very nearly the proportion of one part of oxygen to four parts of nitrogen by weight. Carbonic acid gas, the product of all combustion, exists in the proportion of 3 to 5 parts in 10,000 in the open country. Water in the form of vapor, varies greatly with the temperature and with the exposure of the air to open bodies of water.

In addition to the above, there are generally present, in variable but exceedingly small quantities, ammonia, sulphuretted hydrogen, sulphuric, sulphurous, nitric, and nitrous acids, floating organic and inorganic matter, and local impurities. Air also contains ozone, which is a peculiarly active form of oxygen; and another constituent called *argon* has been discovered.

Oxygen is the most important element of the air so far as both heating and ventilating are concerned. It is the active element in the chemical process of combustion and also in the somewhat similar process which takes place in the respiration of human beings. Taken into the lungs, it acts upon the excess of carbon in the blood, and possibly upon other ingredients, forming chemical compounds which are thrown off in the act of respiration or breathing.

Nitrogen composes the principal bulk of the atmosphere. It exists uniformly diffused with oxygen and carbonic acid gas. This element is practically inert in all processes of combustion or respiration. It is not affected in composition, either by passing through a

furnace during combustion or through the lungs in the process of respiration. Its action is to render the oxygen less active, and to absorb some part of the heat produced by the process of oxidation.

Carbon dioxide gas is of itself only a neutral constituent of the atmosphere, like nitrogen; and—contrary to the general impression—its presence in moderately large quantities (if uncombined with other substances) is neither disagreeable nor especially harmful. Its presence, however, in air provided for respiration, decreases the readiness with which the carbon of the blood unites with the oxygen of the air; and therefore, when present in sufficient quantity, it may cause indirectly, not only serious, but fatal results. The real harm of a vitiated atmosphere, however, is caused by the other constituent gases and by the minute organisms which are produced in the process of respiration. It is known that these other impurities exist in fixed proportion to the amount of carbon dioxide present in an atmosphere vitiated by respiration. Therefore, as the relative proportion of carbon dioxide can easily be determined by experiment, the fixing of a standard limit of the amount in which it may be allowed, also limits the amounts of other impurities which are found in combination with it.

When carbon dioxide is present in excess of 10 parts in 10,000 parts of air, a feeling of weariness and stuffiness, generally accompanied by a headache, will be experienced; while with even 8 parts in 10,000 parts a room would be considered close. For general considerations of ventilation, the limit should be placed at 6 to 7 parts in 10,000, thus allowing an increase of 2 to 3 parts over that present in outdoor air, which may be considered to contain four parts in 10,000 under ordinary conditions. An accurate qualitative and quantitative analysis of air samples can be made only by an experienced chemist.

TABLE I
Quantity of Air Required per Person

Standard Parts of Carbon Dioxide in 10,000 of Air in Room	Air Required per Person	
	Cubic Feet per Minute	Cubic Feet per Hour
5	100	6,000
6	50	3,000
7	33	1,980
8	25	1,500
9	20	1,200
10	16	960

Air Required for Ventilation. The amount of air required to maintain any given standard of purity can very easily be determined, provided we know the amount of carbon dioxide given off in the process of respiration. It has been found by experiment that the average production of carbon dioxide by an adult at rest is about .6 cubic foot per hour. If we assume the proportion of this gas as 4 parts in 10,000 in the external air, and are to allow 6 parts in 10,000 in an occupied room, the gain will be 2 parts in 10,000; or, in other words, there will be $\frac{2}{10,000}=.0002$ cubic foot of carbon dioxide mixed with each cubic foot of fresh air entering the room. Therefore, if one person gives off .6 cubic foot of carbon dioxide per hour, it will require $0.6 \div .0002 = 3,000$ cubic feet of air per hour per person to keep the air in the room at the standard of purity assumed—that is, 6 parts of carbon dioxide in 10,000 of air.

Table I has been computed in this manner, and shows the amount of air which must be introduced for each person to maintain various standards of purity.

While Table I gives the theoretical quantities of air required for different standards of purity, and may be used as a guide, it will be better in actual practice to use quantities which experience has shown to give good results in different types of buildings. In auditoriums where the cubic space per individual is large, and in which the atmosphere is thoroughly fresh before the rooms are occupied, and the occupancy is of only two or three hours' duration, the air-supply may be reduced somewhat from the figures given in Table I.

TABLE II
Air Required for Ventilation of Various Classes of Buildings

Air-supply per Occupant for	Cubic Feet per Minute	Cubic Feet per Hour
Hospitals	40 to 100	2,100 to 4,800
High Schools	30	1,800
Grammar Schools	30	1,800
Theaters and Assembly Halls	25	1,500
Churches	20	1,200

Table II represents good modern practice and may be used with satisfactory results.

When possible, the air-supply to any given room should be based

upon the number of occupants. It sometimes happens, however, that this information is not available, or the character of the room is such that the number of persons occupying it may vary, as in the case of public waiting rooms, toilet rooms, etc. In instances of this kind, the required air-volume may be based upon the number of changes per hour. In using this method, various considerations must be taken into account, such as the use of the room and its condition as to crowding, character of occupants, etc. The data as given in Table III will be found satisfactory for average conditions.

TABLE III
Number of Changes of Air Required in Various Rooms

Use of Room	Changes of Air per Hour
Public Waiting Room	4 to 5
Public Toilets	5 to 6
Coat and Locker Rooms	4 to 5
Museums	3 to 4
Offices, Public	4 to 5
Offices, Private	3 to 4
Public Dining Rooms	4 to 5
Living Rooms	1 to 3
Libraries, Public	4 to 5
Libraries, Private	3 to 4

Force for Moving Air. Air is moved for ventilating purposes in two ways: by expansion due to heating; and by mechanical means. The effect of heat on the air is to increase its volume and therefore lessen its density or weight, so that it tends to rise and is replaced by the colder air below. The available force for moving air obtained in this way is very small, and is quite likely to be overcome by wind or external causes. It will be found in general that the heat used for producing velocity in this manner, when transformed into work in the steam engine, is greatly in excess of that required to produce the same effect by the use of a fan.

Ventilation by mechanical means is performed either by pressure or by suction. The former is used for delivering fresh air into a building, and the latter for removing the foul air from it. By both processes the air is moved without change in temperature, and the force for moving must be sufficient to overcome the effects of wind or changes in outside temperature. Some form of fan is used for this purpose.

Measurements of Velocity. The velocity of air in ventilating ducts and flues is measured directly by an instrument called an ane-

mometer. A common form of this instrument is shown in Fig. **1.** It consists of a series of flat vanes attached to an axis, and a series of dials. The revolution of the axis causes motion of the hands in proportion to the velocity of the air, and the result can be read directly from the dials for any given period.

For approximate results the anemometer may be slowly moved across the opening in either vertical or horizontal parallel lines, so that the readings will be made up of velocities taken from all parts of the opening. For more accurate work, the opening should be divided into a number of squares by means of small twine, and readings taken at the center of each. The mean of these readings will give the average velocity of the air through the entire opening.

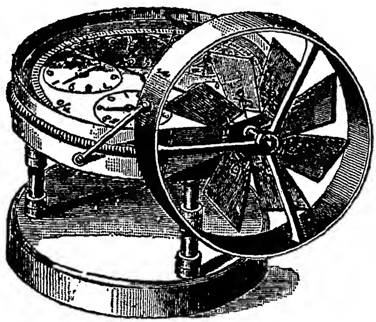


Fig. 1. Common Form of Anemometer, for Measuring Velocity of Air Currents

An anemometer is an exceedingly delicate instrument and great care must be used in handling it. It is suitable only for the measurement of velocities less than 1500 feet per minute. The instrument should be calibrated frequently and its errors corrected, otherwise the results obtained by its use may be very misleading.

Air Distribution. The location of the air inlet to a room depends upon the size of the room and the purpose for which it is used. In the case of living rooms in dwelling-houses where a gravity furnace is used, the registers are placed either in the floor near an inside wall or in an inside wall near the floor; this brings the warm air in at a cooler part of the room. In the case of schoolrooms, where large volumes of warm air at moderate temperatures are required, it is best to discharge it through openings in the wall at a height of 7 or 8 feet from the floor; this gives a more even distribution, as the warmer

air tends to rise and hence spreads uniformly under the ceiling; it then gradually displaces other air, and the room becomes filled with pure air without sensible currents or drafts. The cooler air sinks to the bottom of the room, and can be taken off through ventilating registers placed near the floor. The relative positions of the inlet and outlet are often governed to some extent by the building construction; but, if possible, they should both be located in the same side of the room. Figs. 2, 3, and 4 show common arrangements.

The inlet and vent outlets should always, if possible, be placed in an inside wall; otherwise they will become chilled and the air-flow through them will become sluggish. In theatres and churches

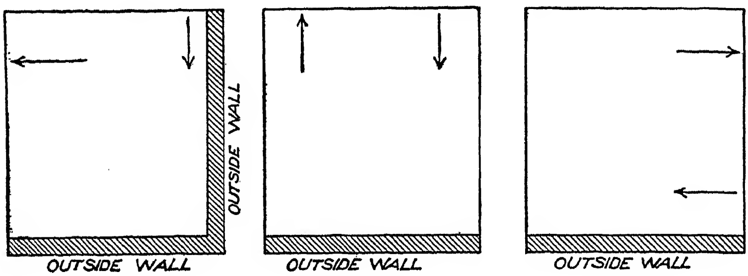


Fig. 2 Fig. 3 Fig. 4

Diagrams Showing Relative Positions of Air Inlets and Outlets as Commonly Arranged

which are closely packed, the air should enter at or near the floor, in finely-divided streams; and the discharge ventilation should be through openings in the ceiling. The reason for this is the large amount of animal heat given off from the bodies of the audience; this causes the air to become still further heated after entering the room, and the tendency is to rise continuously from floor to ceiling, thus carrying away all impurities from respiration as fast as they are given off.

All audience halls in which the occupants are closely seated should be treated in the same manner, when possible. This, however, cannot always be done, as the seats are often made removable so that the floor can be used for other purposes. In cases of this kind, part of the air may be introduced through floor registers placed along the outer aisles, and the remainder by means of wall inlets the same as for schoolrooms. The discharge ventilation should be partly through registers near the floor, or by ceiling vents.

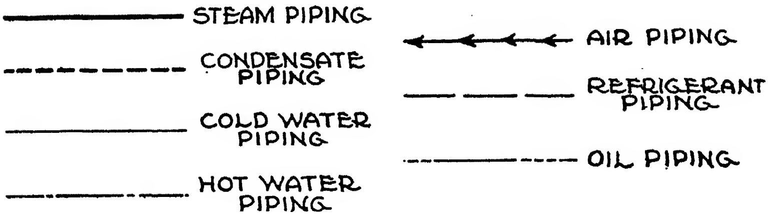


Fig. 5. Piping Symbols

Fig. 6. Automatic Control Symbols

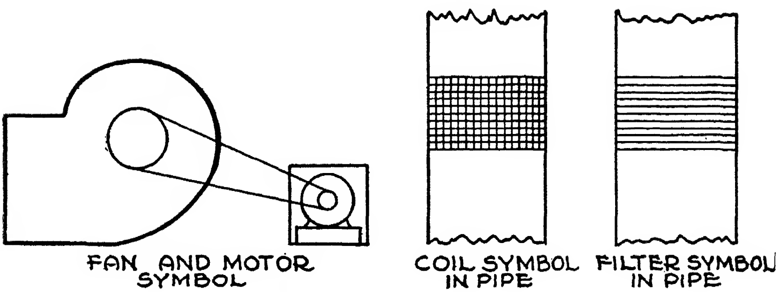


Fig. 7. Miscellaneous Symbols

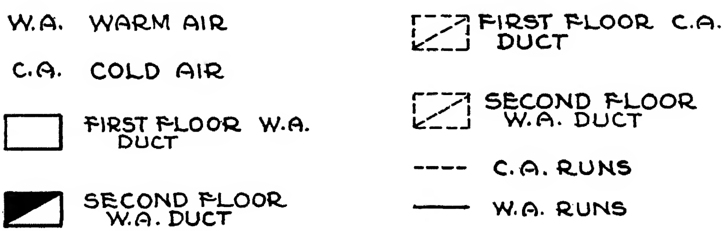


Fig. 8. Duct Symbols

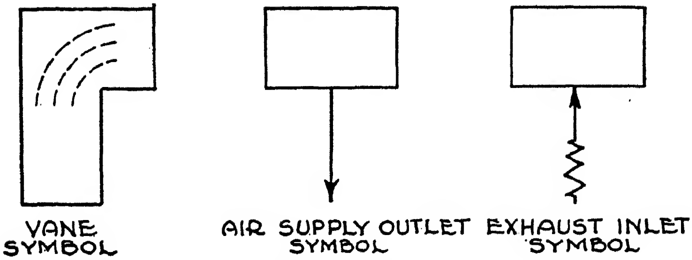


Fig. 9. Duct Symbols

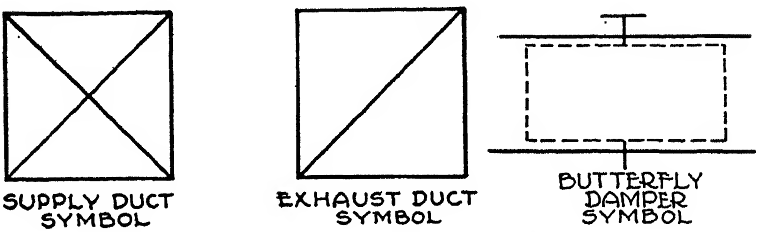


Fig. 10. Duct Symbols

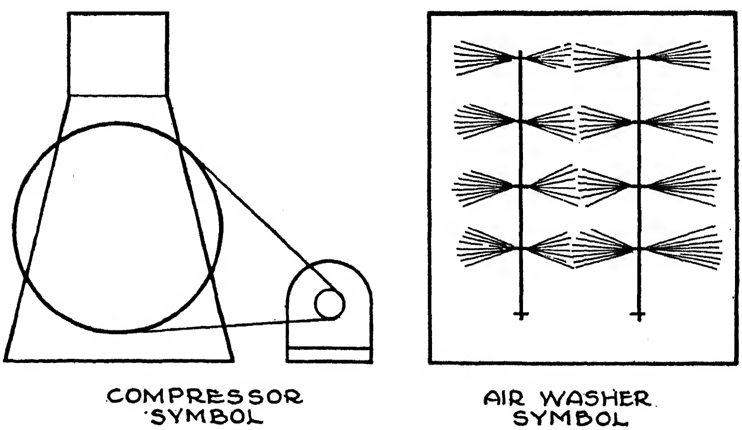
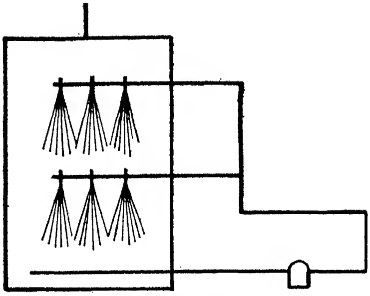


Fig. 11. Equipment Symbols



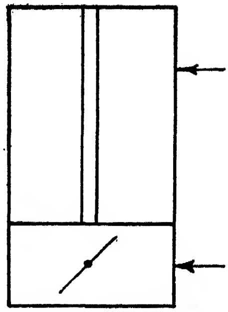


Fig. 12. Equipment Symbols

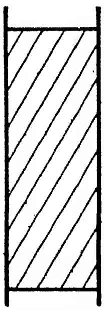


Fig. 13. Equipment Symbol

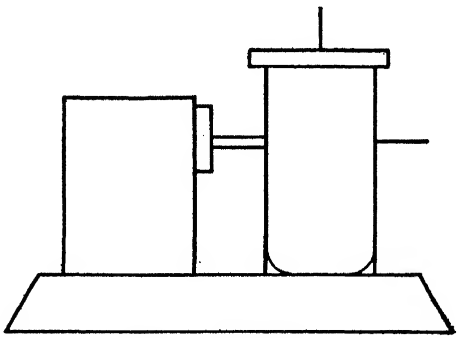


Fig. 14. Equipment Symbol

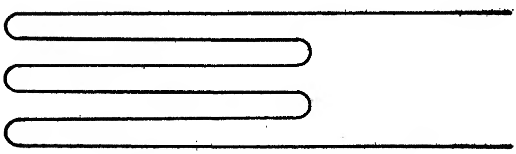
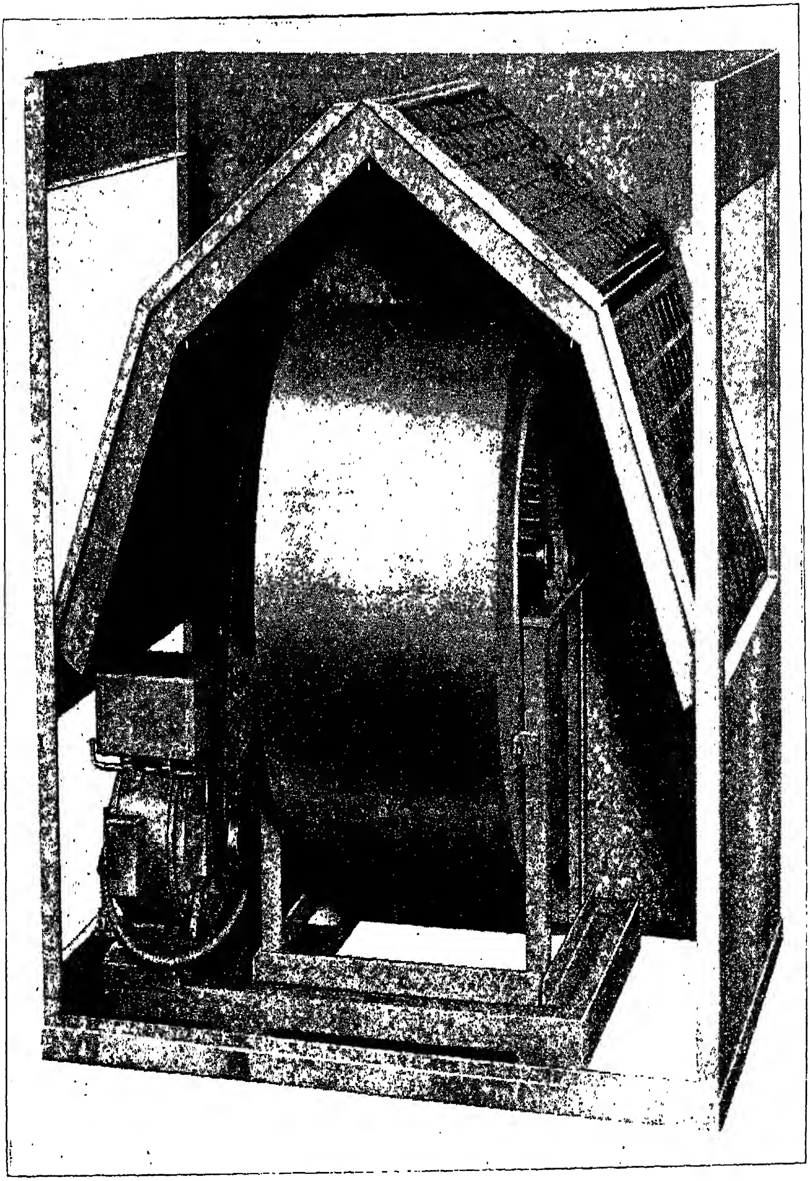


Fig. 15. Equipment Symbol



TYPE OF FAN, MOTOR, FILTER, AND HOUSING ASSEMBLY WHICH CAN BE USED ON NEW WARM AIR FURNACES OR ON CONVERSION JOBS

CHAPTER III

INSULATION

If heating and air conditioning are to be accomplished economically, and at the same time provide the maximum amount of comfort, some means must be found whereby rapid and excessive heat losses or gains can be greatly retarded. Lack of proper consideration of this point results in excessive first costs for apparatus and prohibitive operating expense, especially for air conditioning (cooling) during the warm summer months. Therefore, in this chapter, insulation (the means of retarding heat flow) is discussed and some illustrative examples are given.

Insulation Principles. In this explanation, cork is used because it is typical of all insulations. If a piece of ordinary cork could be examined under a microscope, it would be found to be composed of innumerable very small, cell-like hollow spaces all of which are joined together by thin walls. It could easily be seen that the greater part of the cork is actually composed of these cells. Each cell is sealed, making it a separate "dead-air" space. In other words, no air movement takes place in the cells. Without going into the basic reasons here, it can be explained briefly that in areas (or cells) in which there is no air movement, there can be no heat or cold transfer from one such area to another. Also, because cork is a vegetable growth, even the walls between cells are slow to conduct heat or cold. Therefore, if there is a temperature of 70°F. on one side of a cork partition and 0°F. on the other, the law pertaining to heat traveling from high to low could not normally function, due to the inability of the heat to travel through the cork. It is true that some heat passes through, but the amount is greatly reduced.

Speaking in common terms, it can be said that a light-weight material serves the purpose of an insulator better than a dense material. In this sense, cork is an ideal material when thought of in terms of heat transmission. The difference between a light-weight and a dense material, so far as heat conductivity or transmission is concerned, can be illustrated by assuming an iron rod two feet

long and one inch in diameter, and a wood rod of the same dimensions. Further assume that both rods have one end resting on a very hot stove. After a few minutes the iron rod will be hot throughout its length, whereas the wood will be warm only a distance of perhaps one-third of the way from the end in contact with the heat. This is because iron is very dense (no dead-air cells) and thus offers no resistance to the conductance of heat, whereas wood is much more porous (has a medium amount of dead-air cells) and does offer some resistance to heat conductance. However, wood does not contain enough dead-air cells to make a good insulator for structural purposes.

With cork as a beginning, and used originally only for commercial cold storage rooms and refrigerators, the advancement and development of insulating materials has been rapid. At present there is a wide range of insulators for almost every structural need.

Types of Insulation. *Rigid.* This type, as denoted by its general description, is stiff and is manufactured in sheets of various dimensions and thicknesses. The reason for making it in large sheets is to reduce the number of cracks or joints, to facilitate erection, and because it adds rigidity to a structure. Rigid insulation is made from ground up wood or other fibrous material which has been properly treated and mixed so that when it is formed into sheets its bulk contains millions of dead-air cells. At the same time it has a greater strength than ordinary wood boards of like thickness, and provides much more rigidity than could be obtained using ordinary wood sheathing which it replaces. Rigid insulation can be obtained with a rough surface for ordinary structural purposes, or with a smooth surface for use as a finish material in place of plaster.

Wool. Wool insulation, in appearance, resembles animal wool. However it is generally manufactured from flint-rock, limestone, and other siliceous materials. The manufacturing process produces a soft, gray substance that has the required dead-air spaces in abundance. This insulation has no structural value and must be an addition to a structure rather than a replacement or substitution. It has many uses, especially in insulating structures already built.

Bats. This type is manufactured in blocks or rectangular shaped pads which are generally 15x18 inches in area and have a thickness of approximately 4 inches. The material is wool, as explained for

wool type, covered with a tough paper or wire mesh so that it keeps its shape and is easily applied.

Blanket. Blanket insulation is made of insulating wool as previously explained, or from wood fibres matted together. Both kinds are covered with a tough paper and both have the necessary qualifications to resist heat transfer. Blankets are manufactured in various thicknesses (generally about one inch) and in widths ranging from 17 to 33 inches.

Aluminum Foil. This type of insulation greatly resembles the foil commonly seen on candy or cigars, except that it is aluminum instead of lead or tin. The principle of this form of insulation is entirely different from the dead-air space theory and employs the reflection principle. Its insulating quality lies in its ability to reflect heat. Under normal conditions it reflects approximately 95 per cent of the heat coming in contact with it. Then, when used in multiple layers, it makes use of the spaces between layers, as air spaces, as explained in Chapter V, Vol. II. This material is manufactured in many widths and is of paper thickness.

Quilt. Quilt-type insulation is very much like the previously explained blanket type, except that it is manufactured from a marine growth. This growth is largely composed of silica and has long crinkly fibres, which when matted into the quilt, form millions of dead-air cells. Like other non-rigid types, it must be used in addition to regular structural parts.

Cork. This type is made up into boards and sheets. It is the same material as explained under "Insulation Principles" except that various processes are used in manufacturing, wherein the cork is ground, mixed with cementing materials, and formed into blocks, boards, and sheets.

Ornamental. Ornamental insulation is made from wood or other fibrous materials and formed into designs suitable for trim in a structure. The process is much like that explained for rigid insulation, except that the surface is made smooth and decorative.

Hair Blanket. This type is made from cattle hair. The hair is corded in bats similar to ordinary rolls of cotton, and laid between sheets of laminated paper. It possesses the necessary resistance to heat transfer. It is manufactured in various widths and is about one inch thick.

There are other kinds of insulation but those given are typical.

Where and How Insulation Is Used. *Insulation in Frame Residences—Outside Walls.* Before beginning the discussion of where and how insulation is used, the reader is advised to study Fig. 16 in order to visualize a typical frame wall, such as is encountered in

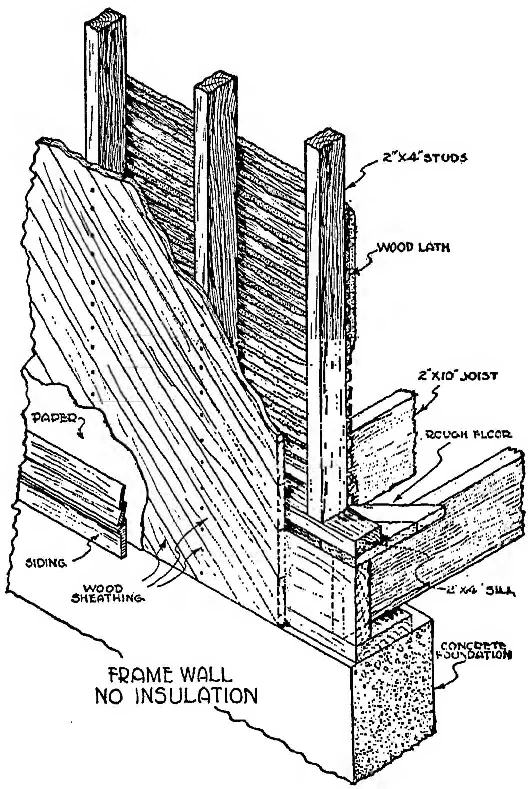


Fig. 16. Typical Frame Wall

an ordinary residence. It is seen that the sheathing is of ordinary wood boards (ship-lapping is sometimes used) nailed directly to the studs. The siding is nailed to the sheathing, and wood laths are also applied directly to the studs. This framing, as explained, is typical of the manner in which construction was carried on in the past. As is shown in Chapter VI, Vol. II, this framing and construction offers little resistance to the conductance of heat, and is very poor economy

even for ordinary old style hot-air heating systems. The plaster keeps out drafts and prevents much leakage but has a poor resistance value, as shown in Table 2, Vol. II, especially when comparing it with Celotex plaster backing shown in Table 14, Vol. II. The wood parts of the frame have poor resistance values as far as pre-

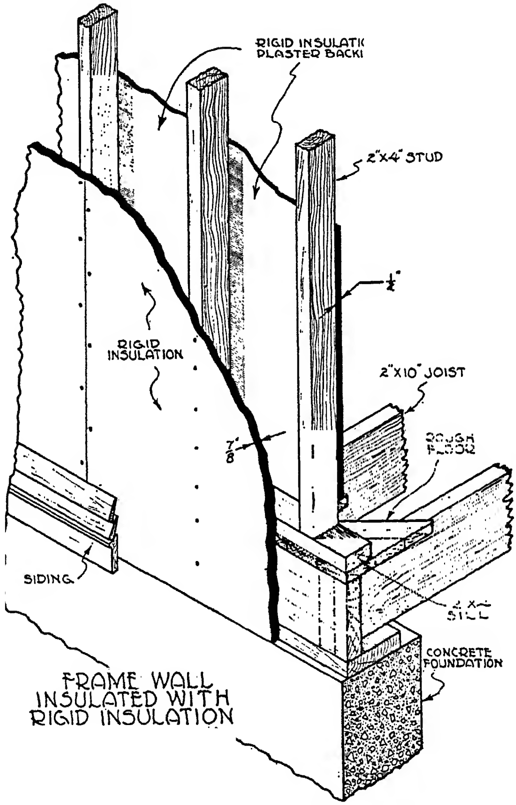


Fig. 17. Frame Wall and Rigid Insulation

venting heat transmission is concerned, so the wall is not at all protected from excessive heat loss or gain.

In Fig. 17 the same type of framing is shown as in Fig. 16. However, Fig. 17 shows the application of rigid insulation in place of wood sheathing and the use of rigid insulation in place of wood lathing. This forms a double means of insulation and at the same time is not unreasonably expensive because wood sheathing and laths

have been eliminated entirely. With this insulation the wall becomes much more resistant to heat transmission, which effects great economy of operation in heating and cooling systems. Also the use of these two sheet forms of insulation makes the entire structure considerably more rigid and eliminates the possibility of plaster crack-

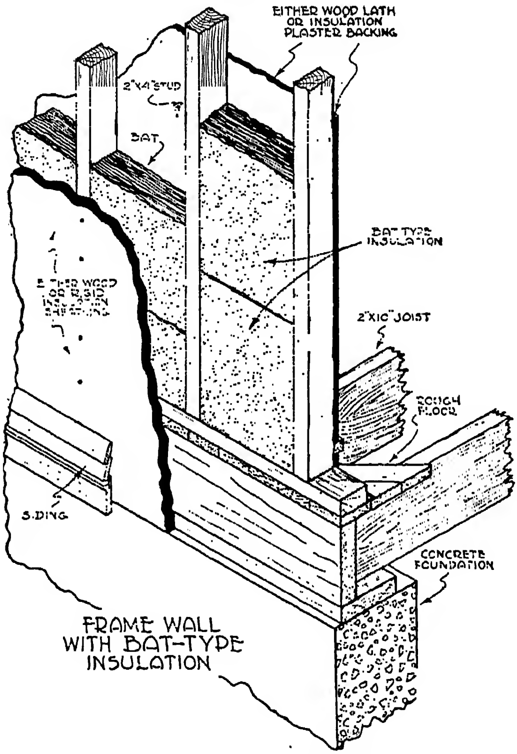


Fig. 18. Frame Wall with Bat and Rigid Types of Insulation

ing. Another less important advantage resulting from such insulation is the tendency to keep the walls clean much longer.

A less expensive, although less effective, method of applying insulation to the wall in Fig. 17 would be to use the rigid insulation only as sheathing and wood laths as plaster backing. This plan is well worth while and saves considerable in fuel alone.

A structural advantage aside from rigidity, is the ease and quickness with which these large sheets of rigid insulation are ap-

plied. Some forms of rigid insulation are manufactured in sizes equal in area to half or one-third the area of one side of a residence. This saves considerable in labor during construction.

Fig. 18 shows three insulation possibilities, any one or all of which may be used with good results. In ordinary construction, not more than one or two are used in the same wall.

The bat type is placed between the studs, where it fits snugly if studs are spaced according to the general practice of 16 inches center to center. They provide practically 4 inches of insulation, which greatly increases the resistance of the wall. The use of rigid insulation together with bats, forms what can be called 100 per cent insulation. While advantageous in many respects, it is too expensive to prove economical within a short period of years. The bats are easily and quickly installed and if care is exercised in placing them they will last as long as the structure and give continuous resistance to heat transmission. It should be pointed out that if the bats are poorly placed, so that gaps appear here and there, the efficiency of the wall as a transmission resistant is greatly lowered. This applies to all wool forms of insulation.

Fig. 19 shows both wool and aluminum foil insulations applied to a wall. The foil is slightly crumpled before being hung. This adds to its reflective ability. To hang the foil, strips of wood or of composition are used. The composition strip is superior because of its thinness and because of the ease in nailing it. The small detail drawing in Fig. 19 shows how the foil is held in position by the strips. The foil may be applied in one, two, three, or more layers, each added layer giving a greater resistance as shown by Table 15, Vol. II. Generally two or three layers are about all that can be applied economically.

Wool insulation cannot be placed until the sheathing and plaster backing are in place. This is because the wool is loose and requires backing to keep it in place. It should be firmly packed so as to leave no voids. Unless wool is firmly packed it will, in time, settle and become compact. This settlement would cause an open space near the top of the wall which would destroy the thermal resistance of the whole wall. Thus exceptional care should be taken to pack the wall firmly but not tightly.

With either foil or wool between the studs, the use of rigid

insulation as sheathing or plaster backing or both is desirable because packing insulation between the studs leaves the studs, plates, sills, etc., unprotected, thus leaving the way open for transmission through their bulk.

Fig. 20 shows the application of blanket or quilt insulation. Both of these materials are flexible, as suggested by their names, and

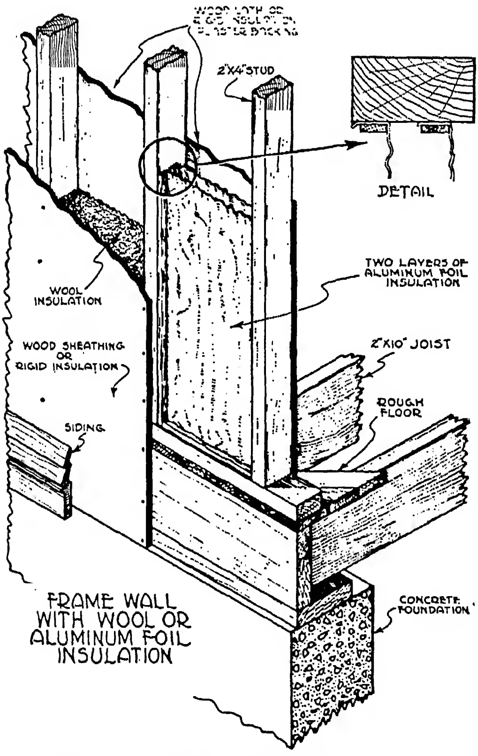


Fig. 19. Frame Wall with Aluminum Foil, Wool, and Rigid Insulation

must be held in place by strips of wood such as laths. Either material is made in widths suitable for installation between studs placed 16 inches center to center. The use of at least one rigid insulation with blankets is advised for the reasons already assigned in the preceding paragraph. In applying either blankets or quilts, care must be taken to nail the holding strip at frequent intervals so that the insulation will hug the studs at all points. It should be kept in

mind that an application of insulation can be likened to water in a receptacle: if there is only one tiny hole, water is bound to leak out.

Insulation in Frame Residence—Partitions. Interior walls or partitions are not usually insulated unless they separate a heated space from an unheated space, or a cooled space from a space not

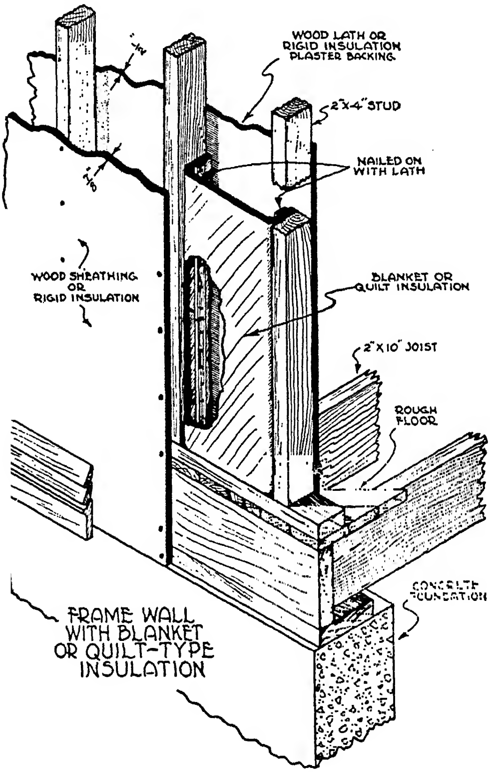


Fig. 20. Frame Wall with Blanket and Rigid Types of Insulation

cooled. For example, the partition in Fig. 122 between the kitchen and basement stairway should be insulated. Or, if only the living room or one bedroom were to be cooled during the summer, it would be wise to insulate the interior partitions surrounding that room. Rigid insulation could be used in place of laths on one or both sides of the partition. Foil or blankets could also be used. Wool is not generally used for interior partitions.

Insulation in Frame Residence—Floors and Ceilings. Fig. 21 shows part of a floor and ceiling. An ordinary floor consists of rough and finish flooring both of wood. To insulate a floor, rigid insulation can be substituted for the rough flooring. This makes an excellent job of insulation and at the same time gives the framing much more rigidity.

The ceiling can be insulated by substituting rigid insulation for the lathing as shown in Fig. 21. The use of rigid insulation for both

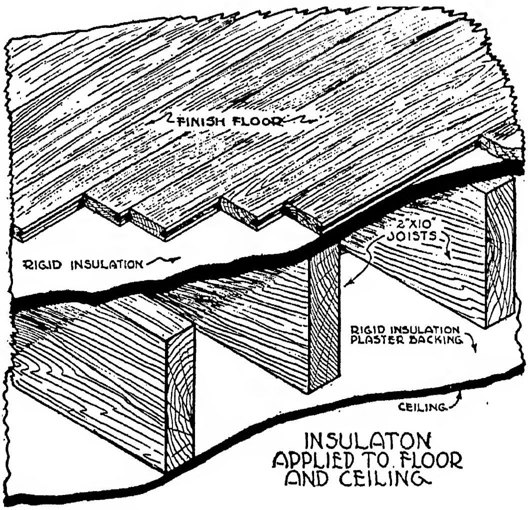


Fig. 21. Use of Rigid Insulation in Floor and Ceiling

floor and ceiling makes a floor and ceiling of high thermal resistance. The ceilings of unheated basements are often covered with rigid insulation without plaster. This gives the necessary thermal resistance and, in addition, gives the basement a finished appearance. Such insulation is often used in attic floors where the attic is unheated.

Wool can be used between the joists instead of the rigid insulation. This, however, requires lath and rough and finish flooring so that no saving in material can be made. At any time after the lathing has been applied, the wool can be packed in, using the same care as explained for walls.

Blankets or quilts can be applied to floors in the same manner as outlined for walls.

Foil is not well adapted to the insulation of floors although it can be used. It is apt to bulge and tear unless extreme care is used in its application.

Insulation in Frame Residences—Roofs. In the ordinary roof the rafters are covered with a wood sheathing upon which are applied

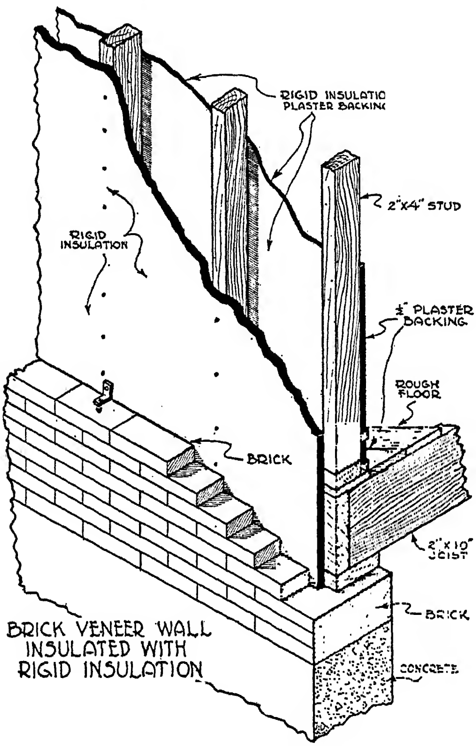


Fig. 22. Typical Brick Veneer Wall with Rigid Insulation

shingles or other roofing. If the attic is unfinished, the rafters are generally bare. Such a roof offers little or no resistance to the high heat imposed on it by the sun. The result is that the rooms beneath the attic become unbearably hot and uncomfortable. The reason for this can more easily be understood when it is known that on a day when the air temperature is 95°F., the temperature at the roof might be as high as 130°F. or 140°F. During the winter, the heat escape through an uninsulated roof is so great as to be a huge waste of fuel.

To insulate a roof, rigid insulation can be substituted for the roof boards or sheathing. The insulation is nailed to the rafters and the roofing applied directly to the insulation or, preferably, to furring strips which are first applied to the insulation. The use of furring strips facilitates laying of the roofing and adds an air space which is of benefit thermally. Rigid insulation can also be applied to the attic

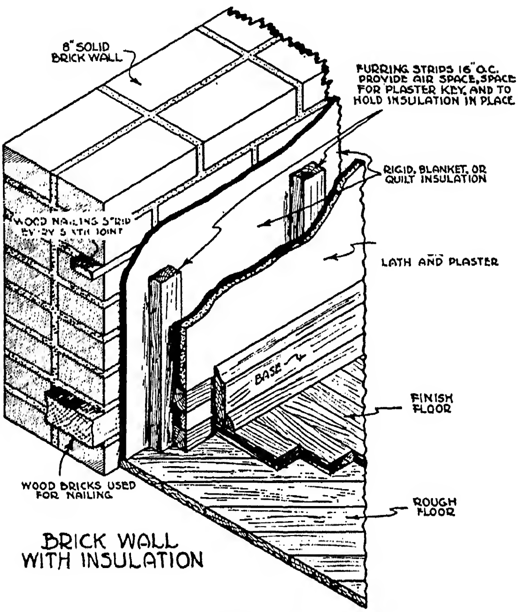


Fig. 23. Brick Wall with Insulation

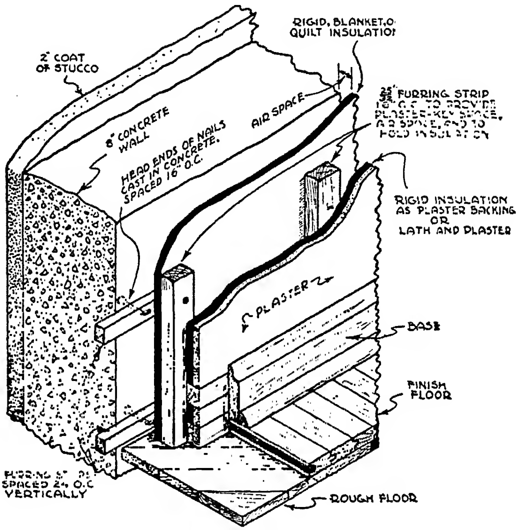
side of the rafters. Foil can be applied between the rafters, using the same method as described for walls.

In designing roof insulation, it is well to use an insulator with small bulk, otherwise heat storage will take place during the summer months, which makes for discomfort at night when the stored heat is released.

Insulation in Brick Veneer Residence. A typical brick veneer wall is shown in Fig. 22. The framing is much the same as that shown in Fig. 16. A veneer wall requires sheathing. To insulate such a wall, rigid insulation can be substituted for wood sheathing. Insulating plaster backing may also be used. Wool, blankets, quilts, and foil can be applied as described for Figs. 19 and 20.

Insulation for interior walls, floors, ceilings, and roofs is the same as has been described for a frame wall.

Insulation in Brick Residence. The insulating of brick walls is more or less confined to the use of rigid, foil, and blanket or quilt-type insulations. Fig. 23 shows a typical brick wall with insulation in place. Rigid, blanket, or quilt insulation can be applied



CONCRETE WALL WITH INSULATION

Fig. 24. Concrete Wall with Insulation

directly to the masonry. Then furring strips are placed over the insulation and either lathing or rigid insulation is applied to the furring strips. To use foil, requires furring strips directly against the masonry. Then the foil is applied between strips.

Insulation in Concrete Residence. Fig. 24 shows a concrete wall. The method of insulation is obvious.

Summary. Not all types of buildings have been pictured or considered in the foregoing explanation, but enough typical examples have been shown to illustrate the general principles.

The reader should keep in mind that insulating a structure is really an engineering procedure and that it requires care and planning. Economically speaking, insulating should pay for itself in fuel and energy savings within a few years' time. A poorly insulated

residence will require much larger apparatus and consume more fuel and energy for year-round conditioning than a well insulated house. This fact alone, substantiates the contention that buildings should be well insulated.

The reader can, by some simple calculations, prove to himself the enormous benefits of insulation in fuel and energy savings. Assume 500 square feet of frame wall such as shown in Fig. 16. Calculate the value of k and compute the heat loss in B.t.u. when the temperature difference is 70°F. Extend this figure over 210 days. Then assume the same wall insulated with the various insulations previously described and again compute heat losses for 210 days. A comparison with the loss for the uninsulated wall shows the great savings. This can be converted to actual fuel savings by the method explained in Chapter VI, Vol. II. Like comparisons can be made by figuring the cooling load and the savings expressed in size of compressors, energy costs, etc.

CHAPTER IV

AIR-CONDITIONING PRINCIPLES

Air conditioning is primarily the control and treatment of air as regards temperature, pressure, water vapor content, velocity, and distribution. The various terms used in air conditioning are explained first, after which the principles of application are given.

Terms. The following terms will be used generally throughout the explanation of Air Conditioning and therefore should be understood thoroughly.

Humidity. Humidity is the amount of water vapor present in a gas. Air is considered as a gas. Humidity is also the number of pounds of liquid vapor carried by one pound of dry gas. This value is sometimes called *absolute humidity.*

Relative Humidity. Relative humidity is the ratio of the weight of water actually contained in a definite volume of gas, to the weight that the same volume of gas is capable of containing when fully saturated, and at the same temperature.

The following definitions will be found of great value for all studies involving humidity, and are substantially those proposed by William Grosvenor. For general use, "air" and "water" will be considered interchangeable with "gas" and "fluids" respectively.

Percentage of Humidity is the number of pounds of liquid vapor carried by one pound of dry gas at a definite temperature, divided by the number of pounds of vapor which one pound of dry gas would carry if completely saturated at the same temperature. Percentage of humidity and per cent relative humidity are not the same, as will be shown later, and a clear understanding of their difference should be gained.

Per Cent Relative Humidity is the percentage ratio existing between the actual partial pressure of gas-liquid mixture and the total partial pressure of the saturated mixture.

Humid Heat is the quantity of heat necessary to raise the temperature of one pound of dry gas plus its contained liquid vapor, one degree Fahrenheit. Since specific heat consists of dry gas, and since

liquid vapors are substantially constant for the temperature range under consideration, the humid heat is calculated from the formula:

$$S = .238 + .48$$

where S is the humid heat and .238 and .48 are the specific heats of air and water vapor respectively.

Humid Volume is the volume of one pound of dry gas together with the liquid vapor it contains. In English units this volume is expressed in cubic feet and is dependent upon the pressure, temperature, and humidity of the mixture.

Saturated Volume is the volume of one pound of dry air together with the amount of liquid vapor it contains when saturated.

Specific Volume is the number of cubic feet occupied by one pound of gas under standard conditions; it increases by its linear dimension with temperature.

Thermometers. Instruments for the measurement of temperature are called thermometers. Two scales of temperature are in use. On the Fahrenheit scale, the boiling point is marked 212°, and the freezing point 32°, there being 180° between them. The Centigrade scale, which is more convenient for scientific work, has its boiling point marked 100° and its freezing point 0°.

We may convert Centigrade degrees to Fahrenheit degrees in the following way: Since 100 Centigrade degrees cover the same temperature interval as 180 Fahrenheit degrees, one Centigrade degree is $\frac{180}{100}$ or $\frac{9}{5}$ as long as one Fahrenheit degree. Hence a temperature of m degrees Centigrade is equal to $\frac{9}{5}m$ on the Fahrenheit scale. But this point is marked 32° on the Fahrenheit scale, consequently the total reading on the Fahrenheit thermometer will be

$$\frac{9}{5}m + 32° \tag{17}$$

The formula for changing a temperature in Centigrade degrees to its Fahrenheit degrees equivalent, is

$$F° = \frac{9}{5}C° + 32° \tag{18}$$

and by transposing we obtain the corresponding formula,

$$C^\circ = \frac{5}{9}(F-32) \tag{19}$$

EXAMPLES FOR PRACTICE

1. To what temperature Fahrenheit does 58°C. correspond?
Ans. 136.4°F.
2. To what temperature Centigrade does 149°F. correspond?
Ans. 65°C.
3. Lead melts at 327°C. What is its melting point on the Fahrenheit scale?
Ans. 621°F.

Dry-Bulb and Wet-Bulb Thermometers. A dry-bulb thermometer is one that is used to measure the temperature of a room or out of doors.

A wet-bulb thermometer has its bulb covered with cloth which must be wet before the temperature is taken, as previously explained. For the same air, a wet-bulb thermometer will give a lower reading than a dry-bulb thermometer; how much lower, will depend on the amount of evaporation from the cloth covering the bulb. The degrees that the wet-bulb temperature is below the dry-bulb temperature is called the depression and indicates the amount of moisture in the air.

How to Find Relative Humidity. This can be done readily by using a sling psychrometer (see Fig. 25), or the Psychrometric Chart in the back of the book. If a sling psychrometer is not available, however, an ordinary dry-bulb thermometer (such as is used ordinarily for temperature readings) will serve the purpose though a little additional work is involved.

To find relative humidity, the dry-bulb temperature of a room, or any other space, is obtained by leaving the dry-bulb thermometer (for a few minutes) in the area to be tested. For the wet-bulb test it is advisable to place a cup of water in the room one day previous to making the test so that the temperature of the water will be the same as that of the area being tested.

After the dry-bulb reading has been taken, tie a small piece of muslin or some similar material around the mercury-well end of the thermometer. Dip this in the cup of water and, taking hold of a string tied to the upper end, swing the thermometer around in the air for about half a minute.

Note the drop in temperature from the original dry-bulb reading. Re-wet the muslin wick and swing the thermometer again for a like period of time to make sure the maximum drop in temperature has taken place. The temperature difference should then be noted. Then, both wet-and-dry-bulb temperatures being known, the Psychrometric Chart in the back of the book can be used to determine the relative humidity. The point of intersection of the vertical line

Fig. 25.

Taylor Sling Psychrometer. The advantage of this form of wet-and-dry-bulb hygrometer over the stationary form is the facility with which tests can be made and the accuracy of the readings obtainable, as in whirling the bulbs they are subjected to perfect circulation. Consists of two accurately etched stem thermometers mounted on a die-cast frame, with the bulb of one covered with a wick to be moistened.

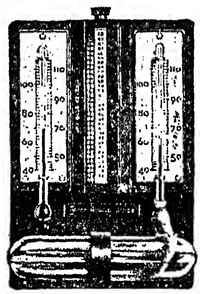


Fig. 26.

Taylor Humidiguide. A handsome small hygrometer for the wall of the home, office, school or other building where a neat, easily read and inexpensive instrument is desired. It is self-contained, requiring no charts or separate tables. Frame is Mahogany Bakelite.

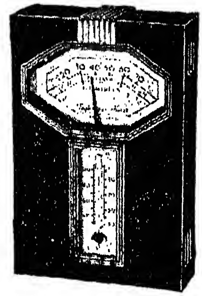


Fig. 27.

Taylor Hampton-Model Humidiguide. (Direct Reading)—A hygrometer giving direct humidity percentages, in a smart, modern case suitable for home, office or public buildings. Finish is satin black with chrome trim.

The Permacolor Thermometer is filled with non-fading red liquid, which is easily read.

Courtesy of Taylor Instrument Company, Rochester, New York

from the dry-bulb temperature on the chart, and the diagonal line from the wet-bulb temperature, gives the relative humidity. For example, turn to the Psychrometric Chart in the back of the book. Assume a dry-bulb temperature of 70°F. and a wet-bulb temperature of 61°F. From the 70 on the bottom of the chart follow the vertical line upward until it intersects the slanting line from 61° on the curved wet-bulb temperature line. They meet at the percentage of humidity line marked 60%. Therefore the relative humidity is

60 per cent. The above method is probably one of the simplest used for this purpose and if carried on carefully, accurate results can be obtained.

There are instruments that give the relative humidity at a single reading such as those illustrated in Figs. 26 and 27. Such instruments can also be depended on for accurate figures.

Dew Point. Dew point is the temperature of air at which any reduction in temperature will cause some of the vapor to be condensed or "squeezed out." This principle is made use of to a great extent in dehumidifying operations. Air at various temperatures can hold definite amounts of moisture. When the air contains the maximum amount of moisture which it can hold at a given temperature, it is said to be "saturated." *For every temperature, there is a corresponding amount or weight of water vapor present.* Now, if air is saturated at a given temperature and this temperature is suddenly lowered, moisture will be "squeezed out" until the air is saturated at the new temperature.

Pressure Measurement. Force as applied to units of area is called pressure and is expressed in pounds per square inch, pounds per foot, inches of mercury, inches of water, atmospheres, etc. Steam gauges read pressures below as above atmospheric pressure. A gauge recording pressure below atmospheric is called a vacuum gauge. The absolute pressure is the sum of the gauge and atmospheric pressures. For example, atmospheric pressure is 14.7. This is often assumed as 15 for ease in calculations. Then absolute pressure is gauge pressure plus 15. Most steam tables give readings in terms of absolute pressure.

Temperature Measurement. Heat energy is understood to lie in the rapid, irregular vibrations of molecules of which all water is composed. The molecules move rapidly from one body to another; or, in other words, heat may flow from one body to another. The conditions that determine which way the flow takes place are called temperature measurement. Equality of temperature may be estimated quite accurately simply by touching two bodies with the hand, provided they are of a similar nature. The power that enables us to do this is called temperature sense.

The Psychrometric Chart in the back of the book represents a new contribution to the Air-Conditioning industry on two major

counts. First, the calculations from which the chart was drawn are based upon more accurate data than heretofore prevailed. Second, the arrangement of the lines makes it possible to obtain desired results directly from the scales on the chart without the use of auxiliary curves or calculations.

The basic properties of air and water vapor used in the calculation of the values from which the curves were drawn were taken from Keenan's Steam Tables and the International Critical Tables. The inclusion of the Specific Volume lines and Total Heat lines on the chart makes it possible to read the numerical values of these items directly for any chosen point on the chart.

Thus, the chart is a fundamental and comprehensive tool for the use of the air-conditioning engineer, and with it he can obtain the complete solution to an air-conditioning problem involving heating, humidifying, cooling and dehumidifying, evaporative cooling, chemical drying, mixing of air at different conditions, preheating, reheating, booster heating, duct losses, or by-passing, or any combination of these.

Use of the Chart. Figs. 28, 29, 30 and 31 are abbreviated Psychrometric Charts and are presented in explanation of the large chart in the back of the book. The curved lines represent the relative humidity.

If two properties of air are known, all properties may be found as follows:

Dry-Bulb Temperature is read directly by following vertically down to the bottom scale.

Wet-Bulb Temperature (Fig. 28) is read directly at the intersection of the wet-bulb line with the 100 per cent relative humidity line (saturation curve). The scale is marked along the 100 per cent line.

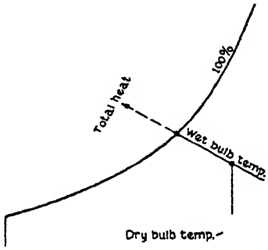
Relative Humidity (Fig. 29) is read directly from the curved lines marked Relative Humidity. For a point between the lines, estimate by distance.

Moisture Content, or Absolute Humidity, (Fig. 30) is read directly from horizontal lines with scales to the right and left of the chart, and is the weight of water vapor contained in a quantity of air and water-vapor mixture which would weigh one pound if all water vapor were extracted.

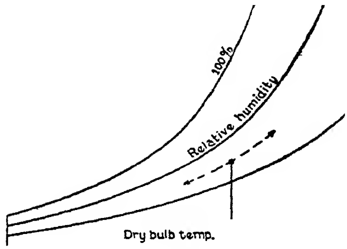
Dew Point Temperature (Fig. 30) is read at the intersection of a horizontal line of given moisture content with the 100 per cent relative humidity line.

Total Heat (Fig. 28) is read directly by following the wet-bulb line to the scale marked Total Heat. Total Heat refers to a quantity of air and water-vapor mixture which would weigh one pound if all water vapor were extracted and includes the heat of the water vapor.

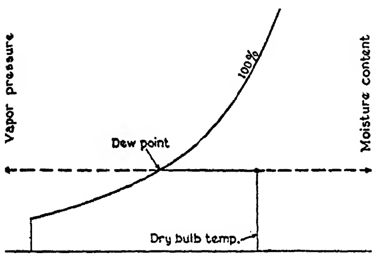
Specific Volume (Fig. 31) is read directly from the lines marked Cu-ft. per Lb. of Dry Air. For points between lines, estimate by distance. Specific volume is the volume occupied by a quantity of air and water-vapor mixture which would weigh 1 lb. if all water vapor were extracted.



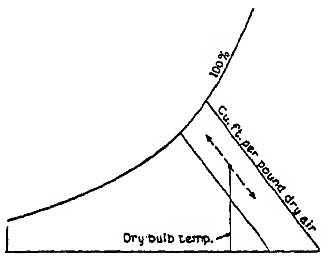
(Fig. 28)



(Fig. 29)



(Fig. 30)



(Fig. 31)

Figs. 28, 29, 30, and 31. Abbreviated Psychrometric Charts
Courtesy of General Electric Company

Vapor Pressure (Fig. 30) corresponding to a given moisture content is read directly from the left-hand scale marked Pressure of Water Vapor.

EXAMPLES

Example 1.* Air at 80° dry-bulb and 65° wet-bulb.
Relative humidity ....45 per cent
Moisture content...................................68.5 gr. per lb.
Dew point...56.8°F.
Total heat (includes heat of 1 lb. of dry air and heat of 68.5 grains of water vapor)...................................30.0 B.t.u. per lb.
Specific volume...................................13.81 cu-ft. per lb.
(Actual weight of 13.81 cu-ft. = 1.0098 lb.; 1 lb. for the dry air and 68.5 grains for the moisture content, 7000 grains = 1 lb.)
Vapor pressure...............................0.228 lb. per sq. in.

**Courtesy of General Electric Company.*

Example 2. Air at 75° wet bulb and 67° dew point.

Dry-bulb temperature.......................................94.7°F.
Relative humidity.......................................40.0 per cent
Moisture content.......................................99.2 gr. per lb.
Total heat.......................................38.5 B.t.u. per lb.
Specific volume.......................................14.28 cu-ft. per lb.
Vapor pressure.......................................0.328 lb. per sq-in.

Example 3. The use of the Psychrometric Chart is very important and in order that the reader may fully understand its use in connection with cooling and dehumidifying, the following typical dehumidifying process is given in detail.

Assume air at 95°F. dry-bulb and 80°F. wet-bulb temperatures. This is represented by point X in Fig. 32.

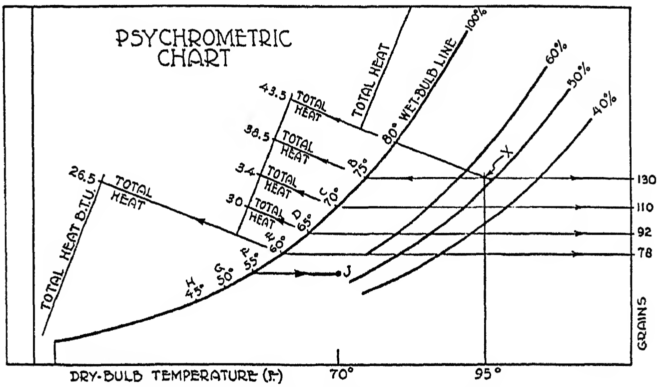


Fig. 32. Abbreviated Psychrometric Chart

Dehumidifying by means of the refrigeration consists of cooling the air below its dew-point temperature. The amount of dehumidification depends on the new dew point to which the air is cooled. If the air at X is cooled to its dew point it is represented by the point B. At this point it has a dew-point temperature of 75°F. but has not lost any of its moisture. In fact at B the air is saturated and has the same moisture content per pound as at X which is 130 grains. Both points X and B are on the same horizontal line for which the moisture content is 130 grains per pound.

If the cooling goes on through points C, D, and E, more and more moisture will be "squeezed out." Thus at point C the moisture content is 110 grains, at D it is 92 grains and at E it is 78 grains per pound. At point E the air would have a temperature of 60°F. and a moisture content of 76 grains, or 54 grains per pound less.

Table 35 gives typical conditions for points X to E in Fig. 32. Items 1, 2, and 3 give the conditions of the air at points X to E. Item 4 gives the sensible heat removed in going from the original point to B, C, D, and E. Item 5 shows the amount of moisture removed subtracting C from B, D from C, etc. Item 6 shows the ratio of the moisture lost to the sensible heat removed. The means of determining the total heat is shown in Fig. 32 by the lines of that name. Thus

if air at X is cooled down to points B, C, D, or E the ratio of moisture to heat removal is shown in Item 6.

Now we can apply the information in Table 35 to a typical example. Suppose in cooling and dehumidifying a given interior space, the total moisture to be removed is 525,000 grains, and the sensible heat, 100,000 B.t.u. per hour*. The ratio would be 5.25. Now if the entering air for the interior space has the condition of point X in Fig. 32, we know that the required dew point is 65°F. for which the ratio is 5.25. The sensible heat between 95°F. and 65°F. is .241 (95–65) which equals 7.23 B.t.u. as shown in Table 35.

Table 35. Data for Example 3

Items	Points	X	B	C	D	E
1	Dry-bulb temperature Wet-bulb temperature	95° 80°	75° 75°	70° 70°	65° 65°	60° 60°
2	Moisture content gr. per lb. of air	130	130	110	92	78
3	Total heat B.t.u. per lb.	43.5	38.5	34	30	26.5
4	Sensible heat removed B.t.u. per lb.*	....	4.82	6.02	7.23	8.43
5	Moisture removed gr. per lb.	....	....	20	38	52
6	Ratio Item 5 / Item 4	....	....	3.32	5.25	6.17

*Use Formula $SH = .241(95° - t)$ where t=temperature at points A, B, C, D or E, and where SH=sensible heat.

To determine the amount of air to dehumidify, the density of the air must be known for 95°F. dry-bulb and 80°F. wet-bulb temperatures. This can be done using Table 30 on the properties of saturated air. Find the column "Volume in Cubic Feet." Then follow the 95°F. line to the right until it intersects the mentioned volume column. It is seen that the volume of one pound of dry air is 13.97 cubic feet and that the volume of one pound of dry air plus vapor to saturate it is 14.79 cubic feet.

$$14.79 - 13.97 = .82 \text{ cubic foot}$$

This is for 100 per cent relative humidity. Fig. 32 shows the air at point X to be approximately 54 per cent relative humidity. Then for 54 per cent, the volume of vapor in one pound of air is $.82 \times 54 = .44$ cubic foot. The volume of one pound of air at 95°F. dry-bulb and 80°F. wet-bulb temperature will, therefore, be $13.97 + .54 = 14.51$ cubic feet. To find the density divide 1 by 14.51 which equals .0689 pounds per cubic foot.

Then

$$\text{c.f.m.} = \frac{100{,}000}{60 \times 7.23 \times .0689} = 3{,}346$$

This equals the quantity of air to be put through the air washer or coils.

Now and then it happens that in order to amply dehumidify (ratio of moisture lost to sensible heat) even more heat than shown in Fig. 32 up to point E or in Table 35 must be removed. Or, in other words, considerably more cooling than represented by point E must be done to remove the necessary amount of

*This process is used for the example given in cooling loads.

moisture. Suppose it is necessary to cool as low as point F. This air at 55°F. would obviously be too cold to use in an enclosure. Therefore it is reheated as shown by line FJ in Fig. 32. The process from X to J has cooled and dehumidified to a comfortable point for places, such as auditoriums where large groups of people gather.

Note 1: In the section on the "Air Washer," additional examples are given, one of which is relative to humidification or just the opposite to the example just considered.

Note 2: The table explaining Mixtures of Air and Saturated Water Vapor in Vol. V together with several basic formulas can be used in place of the Psychrometric Chart in the example just explained. The use of this table and the formulas is explained in the examples given in the section on "Air Washers."

Example 4. It is possible to cool the air by humidification or as it is called, evaporative cooling, although the amount of cooling is limited to the wet-bulb depression and dependent on the efficiency of the air washer. Manufacturers of air washers publish rating tables in which the cooling efficiency of the various washers is given in terms of a per cent of the wet-bulb depression. Cooling efficiency means in other words, the humidifying efficiency of the washer. Air washers, due to lack of efficiency, do not thoroughly saturate the incoming air, although those washers having two banks of sprays reach efficiencies of 90 per cent or slightly better. In general an air washer may be assumed as having an efficiency of 70 per cent.

When the fine spray of water in an air washer evaporates, it requires heat. The process is explained under "Vaporization," page 44. The heat required is latent heat and is taken out of the air passing through the washer. This naturally lowers the temperature of the air somewhat, depending on the wet-bulb temperature of the incoming air. The wet-bulb temperature of the incoming air does not change. This can be seen readily by referring to Fig. 33.

The spray water, if continually recirculated without addition of heat, remains at all times at or very near the wet-bulb temperature of the entering air. Then, since the wet-bulb temperature of the leaving air remains unchanged, it follows that when air is completely (or nearly) saturated it is cooled to the wet-bulb temperature of the incoming air. The efficiency is thus the amount of moisture added to the air compared with the amount required to saturate it adiabatically. This is in direct proportion to the cooling effect.

If an air washer is rated as 70 per cent efficient, then the cooling effect in degrees of temperature that is obtained is 70 per cent of the difference between the original dry-bulb and wet-bulb temperatures.

This can be shown to better advantage by the use of an abbreviated Psychrometric Chart. Fig. 33 shows initial air at 95°F. dry-bulb and 77°F. wet-bulb temperature. This condition is represented on the chart by point X. If the washer were 100 per cent efficient, the line ZX would represent the cooling and humidifying effect. However, if the air washer is only 70 per cent efficient, the point Y represents the dry-bulb temperature of the air leaving the washer. This is calculated as follows.

$$95° - 77° = 18°\text{F.}$$
$$18° \times .70 = 12.6°\text{F.}$$
$$95° - 12.6° = 82.4°\text{F.}$$
$$Y = 82.4°\text{F.}$$

If a line is dropped from point Y, on the chart, to the dry-bulb scale it will indicate a dry-bulb temperature of 82.4°F. Thus a cooling of 12.6°F. is accomplished.

The 12.6°F. is an appreciable amount of cooling. However, the moisture content of the air at point Y is 132 grains or 22 grains above the initial air at point X. Cooling has been accomplished but with added humidity. Many times during warm weather it is impossible to obtain much cooling effect by this method, due to the fact that the wet-bulb temperature is but little lower than the dry-bulb temperature. At times when the wet-bulb temperature is decidedly

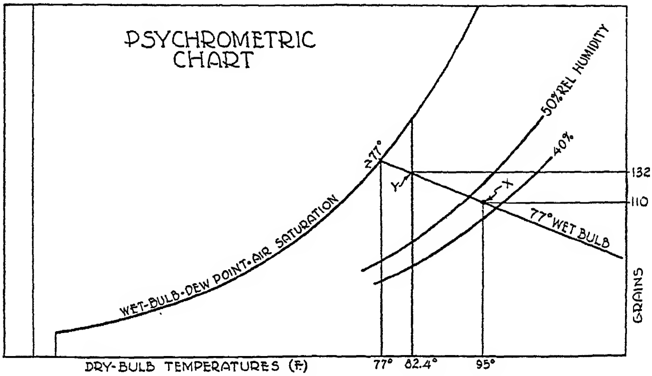


Fig. 33. Abbreviated Psychrometric Chart

lower than the dry-bulb temperature, really appreciable results may be obtained as shown in this example.

***Air Conditioning Processes.** The following explanations together with Figs. 34, 35, 36, 37, 38 and 39, show typical uses of the Psychrometric Chart. For example, dehumidification may be accomplished by passing air over a heat transfer surface, or through a spray of water in an air washer which is maintained at a temperature below the initial dew point of the air. The problem is to so select the surface and its temperature that dehumidification and sensible heat absorption will be in the right proportion to just offset the moisture and sensible heat gains. The relation between dehumidification and sensible heat capacity of a given surface may be determined from a Psychrometric Chart and is illustrated in the following explanations.

Figs. 34 to 39 are abbreviated Psychrometric Charts. The curved lines represent the 100 per cent relative humidity line shown in the Psychrometric Chart found in the back of this book. The points shown in the abbreviated charts can all be located on the big chart.

**Data Courtesy of General Electric Company.*

Sensible Heating and Cooling of Air. (See Fig. 34.) This is represented on the Psychrometric Chart by a straight horizontal line between the dry-bulb temperature limits of the process. These processes are distinguished by a change in dry-bulb temperature, relative humidity, wet-bulb temperature, total heat, specific volume and by no change in moisture content, dew-point temperature and vapor pressure of the air.

Cooling and Dehumidifying of Air. (See Fig. 35.) This is represented on the Psychrometric Chart by a straight line drawn between the initial condition of the air and the point on the 100 per cent line corresponding to the temperature of the cooling surface. This applies only when the surface temperature is below the initial dew point. The final condition of the air will depend on the total heat extracted from the air. This process is distinguished by a change in all properties of the air.

Humidifying of Air. (See Fig. 36.) The humidifying of air with no temperature changes, is represented by a straight vertical line along the dry-bulb temperature line of the air between the moisture content limits of the process. This process is distinguished by an increase in relative humidity, wet-bulb temperature, total heat, specific volume, moisture content, dew-point temperature and vapor pressure of the air.

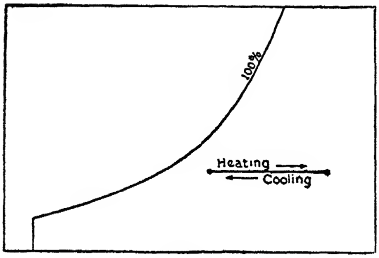
Mixing of Air. (See Fig. 37.) Mixing of air at one condition with air at some other condition is represented by a straight line drawn between the points representing the two air conditions. The condition of the resultant mixture will fall on this line at a point determined by the relative weights of air being mixed.

Evaporative Cooling of Air. (See Fig. 38.) By bringing air in contact with water at a temperature equal to the wet-bulb temperature of the air, is represented by a straight line drawn along the wet-bulb temperature line of the air between the limits of the process. In this process the total heat of the air remains unchanged because the sensible heat extracted from the air is returned as latent heat by an increase of moisture content. This process is distinguished by a change in dry-bulb temperature, relative humidity, specific volume, moisture content, dew-point temperature, vapor pressure and by no change in wet-bulb temperature.

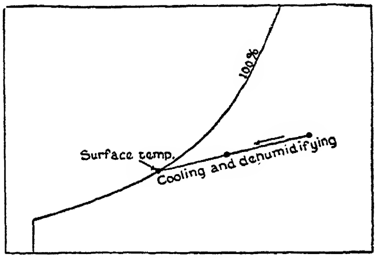
Chemical Drying of Air. (See Fig. 39.) This is represented by a straight line along the wet-bulb temperature between the limits of the process AB only in case the drying is purely by adsorption (the drying agent does not dissolve in the water extracted from the air) and only in case the drying agent does not retain an appreciable amount of the heat of vaporization liberated when the water is condensed on the surface of the adsorber. In case an appreciable amount of this heat is retained by the adsorber, the process takes place on a line below the wet-bulb temperature AB'. If the drying agent is soluble in water (such as calcium chloride) the drying process is above AB'' or below AB' the wet-bulb temperature, depending on whether heat is liberated or absorbed when the agent is dissolved in water.

Measurement of Heat. The measurement of heat in all air-conditioning work, or in all heating work, is generally carried on by the use of the British thermal unit, abbreviated B.t.u., which is the amount of heat required to raise one pound of water from 63°F. to 64°F.

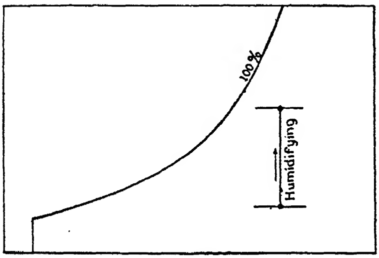
Latent Heat. Latent heat may well be called hidden heat. It is heat stored up in a substance due to changes which take place



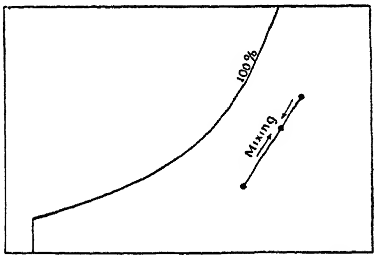
(Fig. 34)



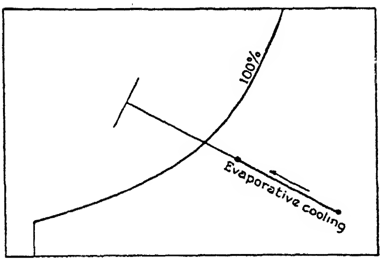
(Fig. 35)



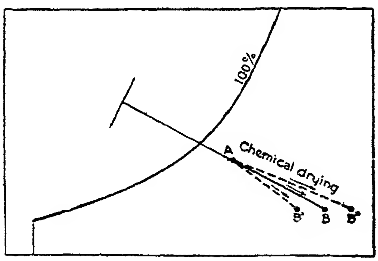
(Fig. 36)



(Fig. 37)



(Fig. 38)



(Fig. 39)

Figs. 34, 35, 36, 37, 38 and 39. Abbreviated Psychrometric Charts

internally. It is heat that does not register on a thermometer or make it self known to our senses.

Latent Heat of Fusion. If ice is heated from, say, 10°F. to 32°F., there is a constant rise of temperature as the heat is added, until the temperature of the ice reaches 32°F. At this point, however, even

though heat is continually added, there is a period over which no rise in temperature takes place. This is the period during which the 32°F. ice is changing to a liquid. Not until every bit of the ice has melted will the temperature of the resulting liquid rise above 32°F. Under such conditions heat is being added to the ice, but for the previously mentioned period no evidence can be seen on the thermometer. This is because the heat is being employed to change the ice from a solid to a liquid without any temperature change. The amount of heat added during the change from a solid to a liquid is called the *Latent Heat of Fusion.* The melting of other substances requires latent heat the same as water.

The latent heat per pound of any substance changed from a solid to a liquid, is a definite amount. For example, to change one pound of ice at 32°F. to water at 32°F. requires 144 B.t.u. Inversely, to make ice at 32°F. from water at 32°F. requires the extraction of 144 B.t.u. per pound.

Table 36 shows latent heats of fusion for several typical substances.

Table 36. Latent Heats of Fusion*

Substance	Heat of Fusion in B.t.u.
Water (ice)	144
Silver	37
Mercury	5
Sea Water (ice)	98
Paraffin	64

*At Atmospheric Pressure.

Latent Heat of Vaporization. The term vaporization means the process of changing a liquid to a vapor. The heat required to make the change from water at 212° F. to vapor (steam) at 212°F. is called latent heat. This heat does not register on the thermometer because boiling water changing to steam remains at 212°F.

The latent heat required per pound of any liquid to change it to a vapor, is a definite amount. For example, water at 212°F. (boiling) requires 970 B.t.u. per pound to change it to vapor (steam). Thus the amount of heat required to change one pound of liquid to a vapor is called its *Latent Heat of Vaporization.*

Table 37 shows latent heats of vaporization for typical substances.

Table 37. Latent Heats of Vaporization*

Substance	Latent Heat of Vaporization in B.t.u.
Water	970
Mercury	122
Alcohol (Ethyl)	370
Sulphur dioxide	172
Carbon dioxide	158
Propane	182

*At Atmospheric Pressure.

Heat is likewise required to evaporate water at ordinary temperatures (below the boiling point). The air-conditioning engineer is more often confronted with evaporation of water at these ordinary temperatures than at 212°F. The following straight-line formula may be used to calculate the latent heat of water vapor for temperatures between 40°F. and 150°F.

$$LH = 1091 - .56t \qquad (20)$$

where

LH = latent heat (B.t.u. per pound)
1091 = a constant
.56 = a constant
t = temperature,°F.

Example. Suppose an air-conditioning system is required to remove moisture from the air of a room when the temperature is 90°F. Calculate the amount of latent heat necessary.

Solution. Substituting in Formula (20),

$$LH = 1091. - .56 \text{x} 90$$
$$LH = 1091 - 50$$
$$LH = 1041$$

Evaporative Cooling. The latent heats of fusion and vaporization have a great value in cooling operations for air-conditioning applications. It has already been pointed out that a pound of water requires a definite amount of heat to change it to a vapor. Thus if air is passed through an air washer in which water is being sprayed in fine mists, some of the water will evaporate (change to a vapor) at the expense of some of the heat in the air. The exact amount of heat used is calculated by using the latent heat Formula (20). By this principle an appreciable cooling effect can be obtained without the use of

artificial refrigeration in air-conditioning systems. Illustrations of this principle in practical use are shown in Example 4, p. 40 and the example, p. 42. The heat of fusion principle is used where ice is employed in an air-conditioning system. One form of air cooling apparatus employs fin-type coils to cool the air passing around them. The coils in turn are supplied with cold water. This cold water is obtained (in systems using ice) by spraying water over pieces of ice in a tank. The water tends to melt the ice and must give up a large share of its heat in doing so. In fact, as previously explained, for each pound of ice melted, the water must supply 144 B.t.u. In this manner the water for the coils is cooled. In some air cooling systems the air is caused to circulate around pieces of ice. In this process, also, the ice requires 144 B.t.u. per pound in order to melt. This heat is extracted from the air and thus the air is cooled.

Specific Heat. The specific heat of air is the ratio between the heat required to raise a given weight of air 1°F. and the heat required to raise a like weight of water 1°F. It is the number of B.t.u. required to raise a pound of air 1°F. The specific heat at constant pressure is generally referred to. The following formula may be used to calculate the specific heat of dry air at constant pressure.

$$S_{cp} = .24112 + .000009t \tag{21}$$

where

S_{cp} = specific heat of dry air at a constant pressure
.24112 = constant
.000009 = constant
t = temperature, °F.

The specific heat of water vapor may be calculated approximately by the following formula.

$$S_{wv} = .4423 + .0001t \tag{22}$$

where

S_{wv} = specific heat of water vapor
.4423 = constant
.0001 = constant
t = temperature, °F.

For ordinary purposes specific heat of air may be assumed as .241 as used by the Buffalo Forge Company in their air-conditioning calculations.

Sensible Heat. Sensible heat is any heat that would be recorded on an ordinary dry-bulb thermometer. The following formula may be used in finding sensible heat.

$$H = S_H(t_2 - t_1) \tag{23}$$

where

H = sensible heat in B.t.u. per pounds
S_H = specific heat (approximately .241)
$t_2 - t_1$ = increase in dry-bulb temperature

Total Heat of Air. Air contains water vapor in amounts depending on temperature and degree of saturation. Thus to find the total heat of air, the latent heat of the vapor is added to the sensible heat. Total heat may be calculated by the following formula.

$$h = 0.24t + Wh_v + .45W(t - t^1) \tag{24}$$

where

h = total heat in 1 pound of dry air plus vapor
$.24$ = constant
t = temperature, water vapor and air
W = weight of vapor in pounds
h_v = heat of vaporization
$.45$ = constant
t^1 = wet-bulb temperature

When air is fully saturated the formula becomes

$$h = 0.24t^1 + Wh_v \tag{25}$$

Adiabatic Saturation. If dry air is combined with water vapor, the temperature of the water is lowered below the temperature of the original dry air. After the combination has taken place, the air is the same temperature as the water. Heat transfer of this kind is called adiabatic saturation of the air.

Superheat. In a refrigerating system, such as one having direct expansion coils, the pressure on the suction side is the pressure in the cooling coils of the evaporator. The vapor is taken from the evaporator by the compressor and compressed to a higher temperature. This work being done on the vapor raises its temperature above the boiling point at its higher pressure. When vapor is in such a condition it is superheated.

Recirculation of Air. The question of whether or not to recirculate air in heating and air-conditioning systems for either residential or larger buildings, is one of great importance, and one that should be given very careful consideration from the economic standpoint.

Take first an ordinary residence. One hundred per cent recirculation of air means that after the air is heated in the furnace it goes up through the leader into the room, out of the room through the cold air register into the cold air return pipe, and eventually back to the furnace again. Thus, recirculation takes place continually.

It is a well-known fact that when recirculation of air is part of a heating system, the air comes back to the furnace at a temperature of between 60°F. and 65°F. Therefore, the furnace does not have to be capable of a very great capacity to heat this 60°F. to 65°F. air up to the necessary bonnet temperature.

Now assume the same residence using only outside air or, in other words, with no recirculation at all. In zero weather the fresh air going into the furnace to be heated enters the furnace at about 0°F. and is heated through a long range of temperatures before it reaches the bonnet temperature. To heat the air through this long temperature range, more fuel and a larger furnace are required than to heat the recirculated air considered first.

The recirculation of air is therefore far more economical for a heating or air-conditioning system, and where filters are used in the furnace, the air is cleaned each time it enters the furnace, and so is as pure as it need be due to this cleaning and humidifying in the winter time. Added to this there is always some infiltration around the windows and doors so that the air in the rooms should be good.

In some systems it is possible to recirculate a certain portion of the air and take in a portion of outside air. The recirculated and outside air are mixed at a mixing point just prior to the point where they enter the furnace for heating. It can be seen that such a system, whereas not quite as economical as the 100 per cent recirculation system, would be more economical than the system using 100 per cent outside air. Such a system can be manually operated, or automatically operated, so that on really cold days the amount of outside air taken in is comparatively small, and on very mild days when the temperature rises to 55°F. or 60°F., almost 100 per cent outside air can be used and very little recirculated air.

The particular system to use on any given installation, therefore depends on many considerations which, as stated, should be carefully thought over before a final design is decided upon.

The system using some outside air, or the system using only outside air can be used to advantage during the summer months to do a little natural cooling. It is a well-known fact that outdoor air generally lowers a few degrees after sunset. The air in a residence, however, due to heat lag of the structural materials, does not lower with the outside temperature, and the result is that the residence is warm and uncomfortable most of the night. Naturally in the summer months the furnace is inoperative, but if a mechanical furnace is being used, the fan can be operated and all of the air taken from the outside. Thus the cool night air is circulated freely throughout the house, which is bound to lower the temperature to about 80 per cent of the difference between the outside and inside temperatures. This has the added effect, if continued all night, of making the house cooler for the first few hours of the next day.

Standards of Comfort. This subject covers a wide field and one in which the opinions of various engineers may vary widely. There are, however, certain factors with which the reader should acquaint himself.

One of the most injurious natural parts of ordinary unconditioned air is dust. The dust content in the air, of course, varies with the locality, depending upon whether it is a manufacturing district, a farm district, etc. The amount of dust in various cities, especially those cities having large manufacturing plants within their borders, is generally much higher than is good for human health. Then there are various kinds of dust coming from such manufacturing processes as those where grinding is going on, which are even more harmful. Such kinds of dust have sharp or pointed edges or corners which are quite invisible to the naked eye, yet very harmful to the lungs.

Soot forms another very injurious and distasteful element in and around cities. The soot enters our lungs, and, of course, is harmful in that way. Soot also is harmful to furniture, even to clothing, and in general is undesirable.

Pollen from various plants at certain times of the year becomes very undesirable to people suffering from various respiratory ailments, and becomes a discomfort to people in perfect health.

Filters, which are discussed elsewhere in this volume, are used extensively to rid the air of the various particles of dust, soot, pollen, and even odors. Air washers, also discussed elsewhere in this volume, are used extensively for the same purpose. Ozone is a form of oxygen molecule, and being unstable in character it is very active as an oxidizing agent. It is used to destroy bacteria in the air, and to offset fumes and odors. The use of ozone is especially desirable in localities where the air has a very strong odor.

Air Motion. Some motion of the air is desirable at all times. Stagnant air, no matter how pure it is, or at what temperature and humidity, is neither comfortable nor healthy. From available data it seems desirable that five feet per minute should be the minimum during the heating season, and fifty feet per minute should be the maximum during the cooling season. These figures have reference to air motion.

The total amount of air circulated in any system is, of course, the sum of the quantity taken from outside, plus the quantity of air recirculated. When considering a cooling and dehumidifying system for summer air-conditioning work, it is desirable to recirculate as great a portion of the air as practical, because of the relatively high cost of cooling. There will always be a considerable quantity of outside air coming into a building by infiltration, which can also be considered.

Effective Temperature. The three factors—temperature, humidity, and air motion—are closely interrelated in their effects upon comfort and health. The atmospheric condition at any time may be such that these various influences will act in opposite directions, and it is therefore the combined or net effect that is considered. This combined or net effect may be expressed as a single value in the form of effective temperature.

The numerical values of the effective temperature scale have been fixed by the temperature of saturated air which induces an identical sensation of warmth. The American Society of Heating and Ventilating Engineers has agreed upon two tables of effective temperatures, based upon dry-bulb temperatures and relative humidity for still air and for persons normally clothed and slightly active. By still air is meant an air movement not in excess of 25 feet per minute. The table showing effective temperatures ranging from 64°F. to 69°F.,

Vol. V, Chapter III, is used when heating or humidification is required. The table for effective temperatures, ranging from 69°F. to 73°F., Vol. V, Chapter III, is used when cooling or dehumidification is required.

The American Society of Heating and Ventilating Engineers has prepared what is called a Comfort Chart, Vol. V, Chapter III, which was plotted with wet-bulb temperatures as the ordinates and dry-bulb temperatures as the abscissas. The straight oblique lines represent relative humidity, and the curved oblique lines represent effective temperatures. The comfort zones are designated by shading on the charts—one for summer and one for winter.

Example. Using the Comfort Chart, Vol. V, Chapter III, and assuming a dry-bulb of 85°F. and a wet-bulb temperature of 67°F, (a measure of relative humidity) the effective temperature is 76°F. Find 85°F. along the bottom line of the chart. Follow this line upwards until it intersects an imaginary line drawn from 67°F. on the left-hand line of the chart. They intersect on the 76°F. effective temperature line. The effective temperature lines are slightly curved.

From this example it can be seen that various combinations of temperature, humidity, and air movement give a composite index which is termed *effective temperature*. The effective temperature is a yard-stick, so called, to measure *effect* produced by heat and cold on a body.

No one set of conditions or combinations of temperature and relative humidity should necessarily be recommended as being ideal, nor can any specific system be considered the best system unless it is based on a comprehensive study of local conditions leading to the desired results within economic limits. Any number of systems can be designated to produce a cooling effect, but only one will be the most flexible and economic to own and operate.

The American Society of Heating and Ventilating Engineers carried out a series of tests in the psychrometric rooms of their research laboratory in Pittsburgh to determine the equivalent conditions met with in general air-conditioning work. The result is shown in a single chart called a Thermometric Chart, Vol. V, Chapter III. The equivalent conditions or effective temperature lines are shown by the short cross lines. The use of this chart can be shown by a typical example.

Example. Given dry-bulb and wet-bulb temperatures of 76°F. and 62°F. respectively, and an air velocity of 100 f.p.m., determine (1) effective temperature of the condition; (2) effective temperature of the still air; (3) cooling produced by the movement of air; (4) velocity necessary to reduce the condition to 66°F. effective temperature.

Solution: (1) Using the Thermometric Chart, draw line *AB* through given dry-bulb and wet-bulb temperatures. Its intersection with the 100-foot velocity curve gives 69°F. for the effective temperature of the condition.

(2) Follow line *AB* to the right to its intersection with the 20 f.p.m. velocity line and see 70.4°F. for effective temperature at this velocity, or so-called still air.

(3) The cooling produced by the movement of the air is 70.4°F. minus 69°F. equals 1.4 degrees.

(4) Follow line *AB* to the left until it crosses the 66°F. effective temperature

line. Interpolating the velocity value is 340 f.p.m., to which the movement of air must be increased for maximum comfort.

Determining Sensible and Latent Heat. This can be explained by the following illustrative problem.

Example*. How much sensible heat, how much latent heat, and how much water vapor will be added per hour to the atmosphere of an assembly hall by an audience of 1,000 grown people when the dry-bulb and wet-bulb temperatures are 75°F. and 63.5°F.?

Solution: Study Curve *D* in Fig. 13, Vol. V. Note that the sensible heat loss per person for a dry-bulb temperature of 75°F. and still air is 265 B.t.u. per hour. From Fig. 14, Vol. V, Curve *D*, it can be seen that the latent heat loss per person for a dry-bulb temperature of 75°F. is 134 B.t.u. per hour, and that the moisture added is 904 grains per hour. Therefore,

$$\text{Sensible heat} = 1{,}000 \times 265 = 265{,}000 \text{ B.t.u. per hour}$$
$$\text{Latent heat} = 1{,}000 \times 134 = 134{,}000 \text{ B.t.u. per hour}$$
$$\text{Water Vapor added} = 1{,}000 \times 905 = 905{,}000 \text{ grains or 129 lbs.}$$

The sensible and latent heat added to the air may also be determined as follows: From Fig. 12, Vol. V, the Thermometric Chart, it can be seen that the effective temperature for dry-bulb and wet-bulb temperatures of 75°F. and 63.5°F. is 70.3°F. Then from Curve *D*, Fig. 15, Vol. V, it can be seen that 403 B.t.u. is the total heat added to the air by a person for an effective temperature of 70.3°F. From Fig. 16, Vol. V, it is found that the percentage of sensible and latent heat for the given temperature of 75°F. dry-bulb is 66.5% and 33.5%. Then—

$$\text{Sensible heat} = 1{,}000 \times .665 \times 403 = 267{,}995 \text{ B.t.u.}$$
$$\text{Latent heat} = 1{,}000 \times .335 \times 403 = 135{,}005 \text{ B.t.u.}$$

Note: The difference between results reached by the two methods is considered negligible.

PRACTICE PROBLEMS

1. Given a dry-bulb temperature of 70°F. and a wet-bulb temperature of 60°F., find the percentage of relative humidity.
2. Find the dew point for the conditions of Problem 1.
3. Given a dry-bulb temperature of 80°F. and a relative humidity of 59%, find the dew point.
4. Find the wet-bulb temperature for the conditions of Problem 3.
5. Given the dry-bulb temperature of 75°F. and a dew-point temperature of 55°F., find the percentage of relative humidity.
6. Find the wet-bulb temperature for the conditions of Problem 5.

*Data Courtesy of American Society of Heating and Ventilating Engineers Guide, 1937.

CHAPTER V

GRAVITY FURNACES

The principles and applications of furnace heating vary widely, depending on heating conditions, temperatures, building construction, geographic locations, economic considerations, and heating re-

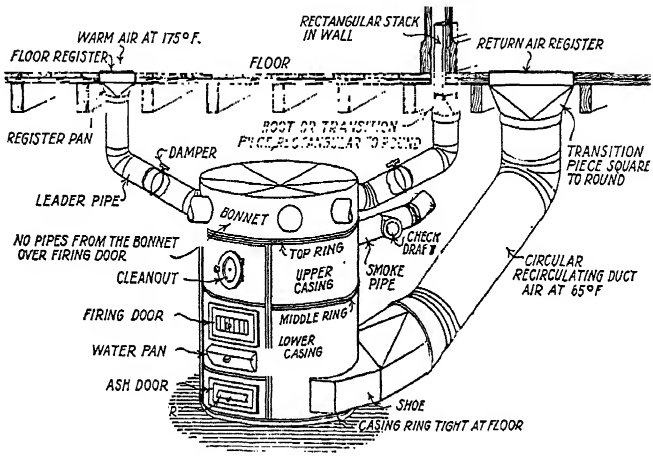


Fig. 40. General Arrangement of Details of a Warm-Air Furnace with Connecting Pipes

quirements. In the following text material, furnaces and typical applications of principles of design are explained for the more common types.

Construction. In construction, a furnace is a large stove with a combustion chamber of ample size over the fire, the whole being inclosed in a casing of sheet iron or brick. The bottom of the casing is provided with a cool-air inlet or shoe, and at the top are pipes which connect with registers placed in the various rooms to be heated. Cold, fresh air may be brought from out of doors through a pipe or duct. This air enters the space between the casings and the furnace near the bottom, and, in passing over the hot surfaces of the fire-pot and combustion chamber, becomes heated. It then rises through the warm-air pipes at the top of the casing, and is discharged through the registers into the rooms above. The air supply should be

brought from the outside only when ventilation is necessary. It is not economical to use an outside air supply in residence heating.

As the warm air is taken from the top of the furnace, cool air flows in through the cold-air duct or recirculating duct to take its

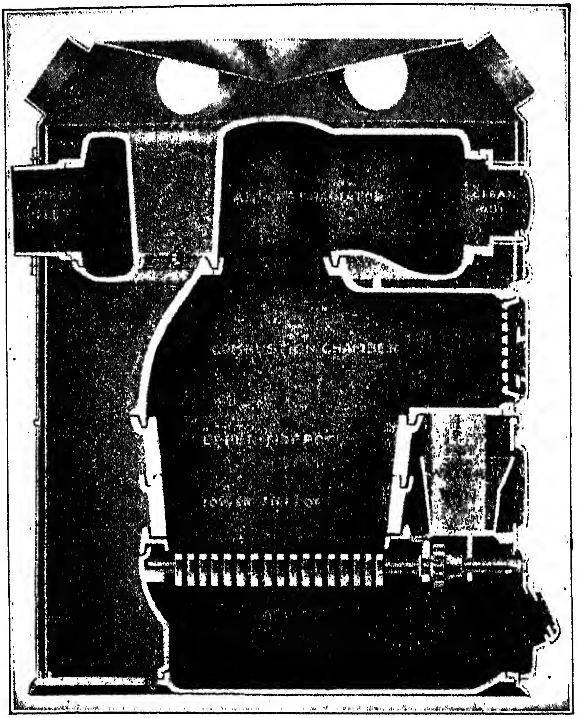


Fig. 41. Section through Direct-Draft Furnace
Courtesy of Fuller-Warren Company, Milwaukee, Wis.

place. The air for heating the rooms does not enter the combustion chamber.

Fig. 40 shows the general arrangement of a furnace with its connecting pipes. The air inlet is seen at the bottom, and the warm-air pipes at the top; these are all provided with dampers for shutting off or regulating the amount of air flowing through them. The feed or fire door is shown at the front, and the ash door beneath it; a waterpan is placed inside the casing, and furnishes some moisture to the warm air before passing into the rooms; water is either poured

into the pan through an opening in the front, provided for this purpose, or is supplied automatically through a pipe.

The fire is regulated by means of a draft damper in the ash door, and a regulating damper placed in the smoke-pipe. Clean-out doors are placed at different points in the casing for the removal of ashes

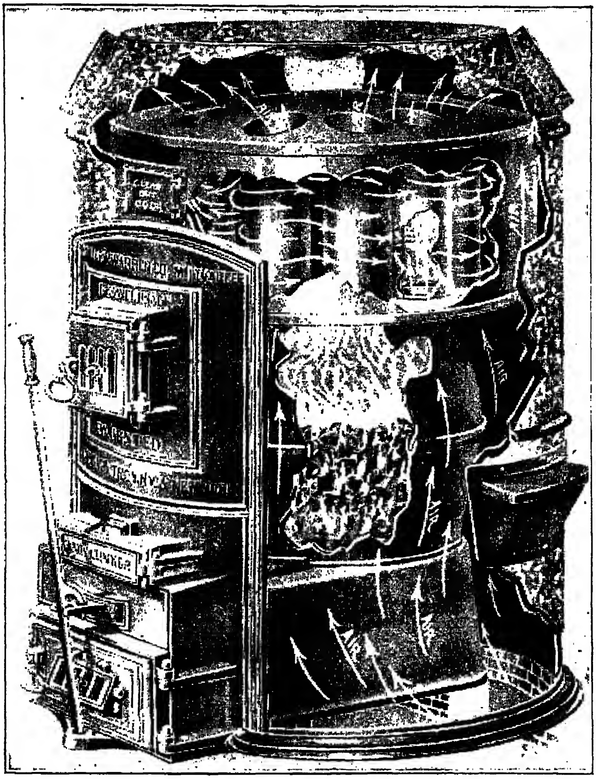


Fig. 42. Stewart "B" Direct-Draft Furnace with Tubular Steel Radiator—Portable Form for Hard or Soft Coal
Courtesy of Fuller-Warren Company, Milwaukee, Wis.

and soot. Furnaces are made either of cast-iron parts cemented together or of steel plates riveted or welded together and provided with brick-lined firepots.

Types. Furnaces may be divided into two general types known as *direct-draft* and *indirect-draft*. Fig. 41 shows in section a common form of direct-draft furnace; the better class has a radiator, gen-

erally placed at the top, through which the gases pass before reaching the smoke-pipe. They have but one damper, usually combined with a cold-air check. Many of the cheaper direct-draft furnaces have no radiator at all, the gases passing directly into the smoke pipe and carrying away much heat that should be utilized.

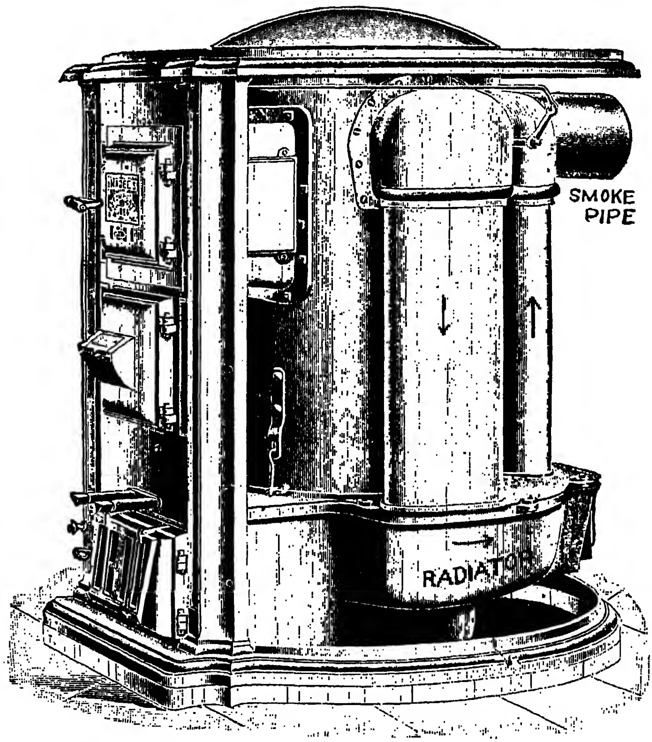


Fig. 43. Indirect-Draft Type of Furnace—Gases Pass Downward to Radiator at Bottom Thence Upward to Smoke-Pipe

The furnace shown in Fig. 41 is made of cast iron and has a large radiator at the top; the smoke connection is shown at the rear.

Fig. 42 represents another form of direct-draft furnace. In this case the radiator is made of sheet-steel plates riveted together with tubular flues passing through it.

In the ordinary indirect-draft type of furnace, Fig. 43, the gases pass downward through flues to a radiator located near the base, thence upward through another flue to the smoke pipe. In addition

to the damper in the smoke pipe, a direct-draft damper is required to give direct connection with the smoke pipe when coal is first put on, to facilitate the escape of gas to the chimney. When the chimney draft is weak, trouble from gas is more likely to be experienced with furnaces of this type than with those having a direct draft.

Grates. No part of a furnace is of more importance than the grates. The plain grate rotating about a center pin was for a long time the one most commonly used. These grates were usually provided with a clinker door for removing any refuse too large to pass between the grate bars. The action of such grates tends to leave a cone of ashes in the center of the fire causing it to burn more freely around the edges. A better form of grate is the revolving triangular pattern, which is now used in many of the leading furnaces. It consists of a series of triangular bars having teeth. The bars are connected by gears, and are turned by means of a detachable lever. If properly used, this grate will cut a slice of ashes and clinkers from under the entire fire with little, if any loss of unconsumed coal.

Firepot. Firepots are generally made of unlined cast iron or of steel plate lined with firebrick. The depth ranges from about 12 to 18 inches. In cast-iron furnaces of the better class the firepot is made very heavy so as to insure durability and to render it less likely to become red-hot. The firepot is usually made in two pieces so as to reduce the liability to cracking. The heating surface is sometimes increased by corrugations, pins, or ribs.

A firebrick lining is necessary in a wrought-iron or steel furnace to protect the thin shell from the intense heat of the fire. Since brick-lined firepots are much less effective than cast-iron in transmitting heat, such furnaces depend to a great extent for their efficiency on the heating surface in the dome and radiator; and this, as a rule, is much greater than in those of cast iron.

Cast-iron furnaces have the advantage when coal is first put on —and the drop flues and radiator are cut out by the direct damper— of still giving off heat from the firepot; while in the case of brick linings very little heat is given off in this way, and the rooms are likely to become somewhat cooled before the fresh coal becomes thoroughly ignited.

Combustion Chamber. The body of the furnace above the firepot, commonly called the *dome* or *feed section,* provides a combustion

chamber. This chamber should be of sufficient size to permit the gases to become thoroughly mixed with the air passing up through the fire or entering through openings provided for the purpose in the feed door. In a well-designed furnace, this space should be somewhat larger than the firepot.

Radiator. The radiator, with which all furnaces of the better class are provided, acts as a sort of reservoir in which the gases are kept in contact with the air passing over the furnace until they have parted with a considerable portion of their heat. Radiators are built of cast iron, of steel plate, or of a combination of the two. The former is more durable and can be made with fewer joints.

The effectiveness of a radiator depends on its form, its heating surface, and the difference between the temperature of the gases and the surrounding air. Owing to the accumulation of soot, the bottom surface becomes practically worthless after the furnace has been in use a short time; surfaces, to be effective, must be self-cleaning.

If the radiator is placed near the bottom of the furnace the gases are surrounded by air at the lowest temperature, which renders the radiator more effective for a given size than if placed near the top and surrounded by warm air. On the other hand, the cold air has a tendency to condense the gases, and the acids thus formed are likely to corrode the iron.

Heating Surface. The different heating surfaces may be described as follows: Firepot surfaces—surfaces acted upon by direct rays of heat from the fire, such as the dome or combustion chamber; gas- or smoke-heated surfaces—such as flues or radiators; and extended surfaces—such as pins or ribs. Surfaces unlike in character and location vary greatly in heating power, so that in making comparisons of different furnaces we must know the kind, form, and location of the heating surfaces as well as the area.

In some furnaces having an unusually large amount of surface, it will be found on inspection that a large part would soon become practically useless from the accumulation of soot. In others a large portion of the surface is lined with firebrick, or is so situated that the air-currents are not likely to strike it.

The ratio of grate to heating surface varies somewhat according to the size of furnace. It may be taken as 1 to 25 in the smaller sizes, and 1 to 15 in the larger.

Smoke Pipes. Furnace smoke pipes range in size from about 8 inches in the smaller sizes to 9 or 10 inches in the larger ones. They are generally made of galvanized iron of No. 24 gauge or heavier. The pipe should be carried to the chimney as directly as possible, avoiding bends which increase the resistance and diminish the draft. Where a smoke pipe passes through a partition, it should be protected by a soapstone or double-perforated metal collar having a diameter at least 8 inches greater than that of the pipe. The top of the smoke pipe should not be placed within 18 inches of unprotected beams. A collar to make tight connection with the chimney should be riveted to the pipe about 5 inches from the end, to prevent the pipe being pushed too far into the flue. Where the pipe is of unusual length, it is well to cover it to prevent loss of heat and the condensation of smoke.

Chimney Flues. Chimney flues, if built of brick, should have walls 8 inches in thickness, unless terra-cotta linings are used, when only 4 inches of brickwork is required. Except in very small houses where an 8 by 8-inch flue may be used, the nominal size of the smoke flue should be at least 8 by 12 inches, and without contractions or offsets. A clean-out door should be placed at the bottom of the flue, for removing ashes and soot. A square flue cannot be reckoned at its full area, as the corners are of little value. To avoid down drafts, the top of the chimney must be carried 2 or 3 feet above the highest point of the roof.

All chimneys and smoke connections should be made tight so that air cannot leak into them. The draft in leaky chimneys and smoke connections is impaired with the result that it may be impossible to burn a sufficient amount of fuel on the furnace grate.

Cold-Air Duct. The cold-air duct should be large enough to supply a volume of air sufficient to fill all the warm-air pipes at the same time. If the supply is too small, the distribution is sure to be unequal, and the cellar will become overheated from lack of air to carry away the heat generated.

The cold-air duct should be as short and direct as possible and preferably circular in cross-section. Care must be taken to avoid all sharp bends and especially sharp right angle turns as the forces producing air flow in a warm-air system are very small. Obstructions to air flow materially cut down the capacity of the system.

If a duct is made too small or is throttled down so that the volume of air entering the furnace is not large enough to fill all the pipes, it will be found that those leading to the less exposed side of the house or to the upper rooms will take the entire supply, and that additional air to supply the deficiency will be drawn down through registers in rooms less favorably situated. It is common practice to make the area of the cold-air duct equal to the combined area of the warm-air pipes. Where air is taken from the outside, the inlet should be placed where the prevailing cold winds will blow into it; this is commonly on the north or west side of the house. If it is placed on the side away from the wind, warm air from the furnace is likely to be drawn out through the cold-air duct.

Whatever may be the location of the entrance to the cold-air duct, changes in the direction of the wind may take place which will bring the inlet on the wrong side of the house. To prevent the possibility of such changes affecting the action of the furnace, the cold-air duct is sometimes provided with check-dampers arranged to prevent back-drafts. These checks should be placed some distance from the entrance so as to prevent their becoming clogged with snow or sleet.

The cold-air box is generally made of matched boards, but galvanized iron is much better. It costs more than wood, but is well worth the extra expense on account of tightness, which keeps the dust and ashes from being drawn into the furnace casing to be discharged through the registers into the rooms above.

The cold-air inlet should be covered with galvanized wire netting with a mesh of at least three-eighths of an inch. The frame to which it is attached should not be smaller than the inside dimensions of the cold-air duct.

Recirculating Duct. It is desirable to recirculate the air from the heated rooms of residences as usually enough air for ventilation is supplied by leakage. The opening or register in the recirculating duct should be placed in the floor in the front hallway of the house so that the cold air from the entrance door will be taken into the furnace. As a rule the stairway of the house is located in the hallway and the return of the air from the upstairs to the furnace is facilitated when the return duct register is located near the foot of the stairway.

Recirculating duct registers located in side walls and under seats are not effective above a height of 6 inches above the floor. Recirculating ducts should be made of galvanized sheet iron and should be circular in section. They should be as short and direct as possible and should equal in area the sum of the areas of the warm-air leader pipes taken from the furnace.

The recirculating duct should not have any sharp bends or any reductions of area and should join into a cold air shoe at the bottom of the casing as shown in Fig. 40. This shoe should be wide and have a height not above that of the furnace grate. This is to prevent the air from being warmed before it enters the furnace casing. Warming the air in the recirculating duct will materially reduce the furnace capacity. The recirculating duct should be insulated from adjacent warm pipes where necessary.

Efficiency. One of the first items to be determined in estimating the heating capacity of a furnace is its efficiency—that is, the proportion of the heat in the coal that may be utilized for warming. The efficiency depends chiefly on the area of the heating surface as compared with the grate, on its character and arrangement, and on the rate of combustion. The usual proportions between grate and heating surface have been stated. The rate of combustion required to maintain a temperature of 70° F. in the house depends, of course, on the outside temperature. In very cold weather a rate of about 8 pounds of coal per square foot of grate per hour must be maintained. These figures should be used in the solution of all heating and ventilating design problems in order to assure accurate and consistent results.

One pound of ordinary good coal will give off about 12,500 B.t.u. and a good furnace should utilize about 55 per cent of this heat. Thus in estimating the required size of a first-class furnace with good chimney draft, we may safely count upon a maximum combustion of 8 pounds of coal per square foot of grate per hour, a B.t.u. value for coal of 12,500 per pound, and 55 per cent furnace efficiency.

Effect of Temperatures on Costs. The average reader has probably never stopped to consider the difference in costs, exclusive of fuel, for heating residences at varying temperatures. To illustrate this point, assume a given residence that requires a total of 2,000 square inches of heat pipe area to heat it to 75°F. when

the outside temperature is $-10°$F. Now, assuming the same residence and a 0°F. temperature outside and 70°F. inside, the total heat pipe required is reduced to approximately 1,500 square inches. This is a difference of 500 square inches of heat pipe area. This brief example serves to show the difference that temperature conditions make in construction costs and indicates the need for careful analysis at all times.

Design of Gravity Systems by B.T.U. Method. *Heating Capacity.* Having determined the heat loss from a building (as in factory building in Vol. II, Chapter VI), it is a simple matter to compute the size of grate necessary to burn a sufficient quantity of coal to furnish the amount of heat required for warming.

In computing the size of furnace, it is customary to consider the whole house as a single room, with four outside walls and a cold attic. The heat losses by conduction and leakage are computed. The heat delivered to the various rooms may be considered as being made up of two parts—*first,* that required to warm the outside air up to 70°F. (the temperature of the rooms); and *second,* the quantity which must be added to this to offset the loss by conduction and leakage. Air is usually delivered through the registers at a temperature of 175°F., or less.

Table 38. Firepot Dimensions

Firepot Diameter Inches	Diameter of Grate Inches	Grate Area Square Feet
22	18	1.77
24	20	2.18
26	22	2.64
28	24	3.14
30	26	3.69
32	28	4.27
34	30	4.91
36	32	5.58

If the air is recirculated, it may be considered as returning to the furnace at 65°F., when 70°F. is being maintained at the breathing line in the rooms heated. The heat in the air available for heating is that obtained by dropping the warm-air temperature in the room from 175°F., the register temperature, to 70°F., the room temperature. However, there are heat losses between the furnace casing and the rooms to be heated so that to maintain 175°F. air temperature at the registers, it is necessary to have an air tempera-

ture of at least 30 degrees F. more at the top of the furnace casing bonnet. In the case under consideration it would be necessary to raise the recirculated air from 65°F. to 205°F. in the furnace casing. The following example may be used as a typical solution of the size of a furnace grate required.

Example. The loss through the walls and windows of a building is found to be 80,000 B.t.u. per hour in zero weather. What will be the size of a furnace required to maintain an inside temperature of 70°F.?

The specific heat of air at ordinary temperatures may be taken as 0.24 B.t.u. per pound. The specific heat of air is the heat in B.t.u. required to raise the temperature of 1 pound of air 1°F.

The temperature drop of the air in heating the room is $175-70=105$°F. The heat given up per pound of air is $0.24\times105=25.2$ B.t.u. The weight of air required for heating is $80{,}000\div25.2=3{,}175$ pounds per hour. The heat that the furnace must supply to this air per hour is computed as follows:

Temperature rise of the air in the furnace casing is $175+30$	$=205-65=140$°F.
Heat added per pound of air	$=0.24\times140=33.6$ B.t.u.
Heat required for the furnace per hour	$=3{,}175\times33.6=106{,}680$ B.t.u.

When the value of the fuel is taken as 12,500 B.t.u. per pound, the furnace efficiency as 55 per cent, and the combustion rate as 8 pounds of coal per square foot of grate per hour, the heat from the fuel available per square foot of grate surface per hour is

$$12{,}500\times.55\times8=55{,}000 \text{ B.t.u.}$$

The grate surface required is

$$106{,}680\div55{,}000=1.94 \text{ square feet}$$

The nearest grate diameter in Table 38 is 20 inches. This would require a 24-inch furnace, as furnaces are rated on the largest diameter of the firepot, which usually is 4 inches larger than the grate diameter.

If the air were taken from the outside instead of the inside, the calculation would be as follows:

Heat added per pound of air	$=0.24\times205=49.2$ B.t.u.
Heat to be supplied by the furnace	$=3175\times49.2=156{,}210$ B.t.u.
Grate area required	$=156{,}210\div55{,}000=2.84$ square feet

Thus a very much larger furnace would be necessary. It can be seen from these calculations that when the air from the house is not recirculated, it requires both a larger furnace and more fuel to heat the house.

Warm-Air Leader Pipes. The required size of the warm-air pipe to a room depends upon the heat losses from the room. A satisfactory method to proportion them is given in Table 39. It is based on the data from Bulletin 141, Engineering Experiment

Table 39. Heat Available above 70°F. at Register in B.t.u. per Square Inch of Leader Pipe

Register Temp. Degrees F.	1st Floor	2nd Floor	3rd Floor
130	36	76	95
140	50	97	120
150	66	119	144
160	81	140	169
170	95	160	195
175	103	170	206
180	112	182	220
190	125	202	243
200	140	223	268

Station, University of Illinois, by Professors A. C. Willard, A. P. Kratz, and V. S. Day.

Example. A second floor room has a heat loss of 8,500 B.t.u. per hour. The room is to be maintained at 70°F. with a register temperature of 175°F. What will be the size of leader required? Use B.t.u. method.

The heat-carrying capacity of a leader to a second-floor room with a register temperature of 175°F. is 170 B.t.u. per square inch, Table 39.

$$\text{Leader area} = 8{,}500 \div 170 = 50 \text{ square inches}$$

Table 40 gives an 8-inch pipe as having an area of 50 square inches. Leader pipes smaller than 8 inches in diameter should never be used, irrespective of what the calculated size may be, as they are liable to be unsuccessful because of their friction and heat losses. Leader pipes should always be carried full size and circular in section. They should never be ovaled to fit to the furnace casing bonnet as the cross-sectional area of the leader is reduced at an ovaled section. Leader pipes should not be greater than 12 feet in length. All leader pipes should grade upward uniformly 1 inch in 1 foot.

Table 40. Dimensions of Warm-Air Leader Pipes

Diameter of Pipe Inches	Area Square Inches	Area Square Feet
8	50	.349
9	64	.442
10	79	.545
11	95	.660
12	113	.785
13	133	.922
14	154	1.07
15	177	1.23
16	201	1.40

Warm-Air Stacks. Leaders serving second floor and third floor rooms are joined to rectangular ducts that go up through inside walls to the rooms to be heated. The connecting piece between the circular

pipe and the rectangular duct is known as a *boot*. Good and bad types of boots are shown in Fig. 44. The stacks should always be run in an inside wall and never in an outside or exposed wall as the results will be unsatisfactory. The cross-sectional area of the stack should equal the cross-sectional area of the leader serving it. This is sometimes difficult to do in frame wall construction. The stack cross-sectional area may be reduced to ¾ that of the leader area with some reduction in capacity, but it should never be made less than ¾ the leader area. Thus an 8-inch leader should not be served by a

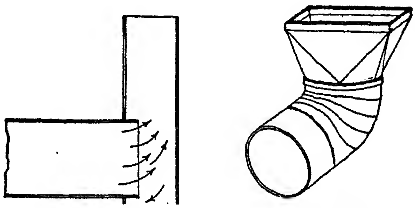


Fig. 44. Good and Bad Types of Boots

stack smaller than 50×0.75, or 37.5 square inches inside. A suitable stack would be 4 inches × 10 inches, or 40 square inches cross-sectional area.

To secure the proper size of the stack, it may be necessary to have the wall studding made of 6-inch material instead of the customary 4-inch studding. Each room should be heated by a separate leader and stack. Large rooms may require more than one leader. The stacks should never be less than 3½ inches in depth, and the breadth should not be greater than three times the depth.

PRACTICE PROBLEMS

1. A first-floor room has a computed loss of 20,000 B.t.u. per hour. The air for warming is to enter through two pipes of equal size and at a temperature of 150°F. at the registers. What will be the required diameter of the pipes? The room is to be maintained at 70°F. Use B.t.u. method.

Ans. 14 inches diameter each.

2. If in the above problem the room had been on the second floor and the air were to be delivered through a single pipe, what diameter would be required? Use B.t.u. method. *Ans*. 15 inches

3. A house having a furnace with a firepot 22 inches in diameter is not sufficiently warmed, and it is decided to install a new furnace. The heat loss

from the building is 153,600 B.t.u. per hour in zero weather. What diameter of firepot should be used in the new furnace? Assume the heat value of fuel as 12,500 B.t.u. per pound, the furnace efficiency as 55 per cent, and the combustion as 8 pounds of coal per square foot of grate per hour. The temperature in the house should be 70°F. Assume recirculated air.

4. A building has a total heat loss of 175,600 B.t.u. per hour in + 10°F. weather. What diameter of firepot should be used? Assume heat value of coal at 11,500 B.t.u. per pound, the furnace efficiency as 65 per cent, and the combustion rate as 7 pounds of coal per square foot of grate per hour. The temperature in the house is 70°F. Assume recirculated air.

5. If all outside air is used for the building in Problem 4, how much larger will the firepot have to be?

Gravity Warm-Air Furnace with Filters. Recently, gravity warm-air heating, having the advantages of filtered air, has been made possible by a furnace like the one shown in Fig. 45.

This type of furnace employs filters at the base, extending all around the furnace, except for the space taken by the firing door, ash door, etc. No cold-air pipes are attached to the furnace. The cold air is drawn from the basement through the filters and then is heated, etc., in the regular manner. The cold-air pipes, instead of being attached to the furnace, are brought down to within 18 inches of the basement floor (see Fig. 50) and left open. Thus cold air returning through the cold-air pipes enters the basement near the floor level and travels over the floor to the filters and into the furnace, after being cleaned by the filters. (Filters are explained on p. 151.)

This system requires less area in cold-air pipes, although the designing rules already given are followed if possible. In the clean and well-kept basements of the present day, this furnace operates well and gives the added convenience of filtered or clean air.

Summary. In the foregoing pages, gravity furnaces have been explained as to construction, various parts, heating capacity, and design of leaders, stacks, and cold-air pipes. That part of the explanation describing parts, such as bonnet, grates, leaders, etc., applies not only to their use on gravity furnaces but also explains the general principles of such parts used in other types of furnaces. For this reason explanations of such parts are not given with the descriptions of other types of furnaces.

The method used to calculate the furnace size and sizes of leaders and cold-air pipes may be called the B.t.u. method because

heat losses, etc., are actually calculated for each room in terms of B.t.u.'s. This method is very accurate and should be employed especially in cases where the residence or building being considered

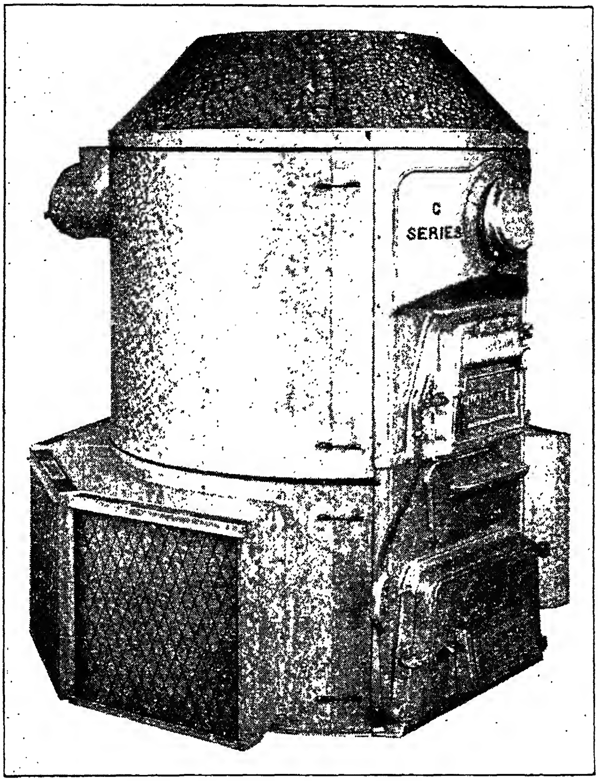


Fig. 45. Moncrief Warm-Air Furnace with Filters
Courtesy of The Henry Furnace and Foundry Company, Cleveland

is at all out of the ordinary in construction or if conditions pertaining to it are unusual.

Note: The method to use in calculating heat losses is covered by the material in Chapter IX.

There is another method of designing furnace systems, which is simpler than the B.t.u. method and which gives results almost as accurate. This method is explained in the following pages. The reader should take special care to learn both methods, how to use them, and differentiate between them.

* **Design of Gravity Systems by Code Method.** The National Warm Air Heating and Air Conditioning Association Code gives a recognized method for designing hot-air gravity systems which is somewhat different from the method previously studied. It is based on long experience, together with many actual trials and experiments and is applied by means of rules and formulas which, while somewhat empirical, are amply accurate for systems for average residence requirements and conditions.

The Code method requires that basement pipes be designed first, with stacks, registers, cold-air pipes, and furnace size following in the order named.

Design of Basement Warm-Air Pipes. The following formula gives the cross-sectional area of basement pipes.

$$A = \left(\frac{G}{12} + \frac{W}{f} + \frac{C}{f} + \frac{F}{f} + \frac{V}{800}\right) \times K \qquad \textbf{(26)}$$

where

A=cross-sectional area of basement pipe to each room in square inches.

G=nominal glass area in square feet, including window or door opening.

W=net outside wall area in square feet (doors and windows deducted).

C=ceiling area adjacent to unheated space.

F=floor area over unheated space.

f=factors taken from Table 41 for the wall, ceiling, and floor constructions.

V=volume of air renewal per hour=volume of room $\times$ number of air changes.

800=a constant.

K=a constant which varies as follows:

For first-floor rooms, K=9

For second-floor rooms, K=6

For third-floor rooms, K=5

Note 1. The reader will note that in this method no actual heat-loss calculations are necessary. As previously explained, this method depends on the results of many trials for the accuracy of results. It can be used successfully in most residence work where no special or out of ordinary conditions must be met. The B.t.u. method, previously explained, should be used for all cases where special conditions exist. Otherwise both methods serve equally well for furnace design.

Note 2. The Code formula is based on 70 degrees temperature difference (outside temperature 0°F., inside temperature 70°F.) When temperature difference is more than 70 degrees, add 1½ per cent per degree above 70 degrees to final figures. When temperature difference is less than 70 degrees, deduct 1½ per cent per degree below 70 degrees from final figures. (The final figures are represented by A in the formula.)

Note 3. If any room has an unprotected north or northwest exposure, add 15 per cent to the calculated pipe areas.

Note 4. Use no basement hot-air pipe less than 8 inches in diameter.

Note 5. Basement pipes should be comparatively straight and should not be over 12 feet in length. Sharp turns and long pipes should have extra

* Data Courtesy of National Warm Air Heating and Air Conditioning Association.

capacity. When pipes exceed 12 feet in length or have more than two 90° turns, the next larger pipe size should be used.

Table 41. Values of *f* for Use in Calculating Furnace-Pipe Sizes for Gravity and Mechanical Systems.

	f
Exposed Wall:	
No. 1*a*. Frame wall constructed of siding, paper, sheathing, studding, lath, and plaster	60
b. Same as (1*a*) construction substituting ½-in. fibrous board or equivalent for the lath	80
c. Same as (1*a*) construction with additional 3½-in. insulating fill between studding	140
For stucco on frame walls, use the same values as for frame with siding, as shown in (1*a*), (1*b*) and (1*c*).	
No. 2. 9-in. brick wall plastered on one side	40
No. 3*a*. 9-in. brick wall, air space, furred, and plastered	57
b. Same as (3*a*) construction substituting ½-in. fibrous board or equivalent for the lath	84
No. 4. 13-in. brick wall, plastered on one side	52
No. 5*a*. 13-in. brick wall, air space, furred and plastered	69
b. Same as (5*a*) construction substituting ½-in. fibrous board or equivalent for the lath	97
No. 6. 4-in. brick, 4- or 8-in. hollow tile plastered	57
No. 7*a*. 4-in. brick, paper, sheathing, studding, lath, and plaster (brick veneer)	58
b. Same as (7*a*) construction substituting ½-in. fibrous board or equivalent for the lath	84
c. Same as (7*a*) construction with additional 3½-in. insulating fill between studding	158
No. 8. Stucco on 8-in. hollow tile, and plaster	48
No. 9*a*. Stucco on 8-in. hollow tile, furred, and plastered	65
b. Same as (9*a*) construction substituting ½-in. fibrous board or equivalent for the lath	95
Ceilings—with attic space above:	
No. 10*a*. Lath and plaster without floor above	50
b. Same as (10*a*) construction substituting ½-in. fibrous board or equivalent for the lath	70
c. Same as (10*a*) construction with additional ½-in. fibrous board or equivalent nailed on top of joists	90
d. Same as (10*a*) construction with additional 3½-in. insulating fill between joists	150
No. 11*a*. Lath and plaster with tight floor above	90
b. Same as (11*a*) construction substituting ½-in. fibrous board or equivalent for the lath	104
c. Same as (11*a*) construction with additional 3½-in. insulating fill between joists	183

No. 12*a*. Metal without floor above.................................. 40

b. Same as (12*a*) construction with additional ½-in. fibrous board or equivalent between metal and joists.......................... 65

c. Same as (12*a*) construction with additional ½-in. fibrous board fastened on top of joists.. 85

d. Same as (12*a*) construction with additional 3½-in. insulating fill between joists...145

No. 13*a*. Metal with tight floor above................................ 75

b. Same as (13*a*) construction with additional ½-in. fibrous board between metal and joists.. 95

c. Same as (13*a*) construction with additional 3½-in. insulating fill 176

Ceilings—without attic space above—part of the roof:

No. 14*a*. Lath, plaster, rafter, sheathing, any type of shingles or roofing 57

b. Same as (14*a*) construction substituting ½-in. fibrous board or equivalent for the lath... 74

c. Same as (14*a*) construction with additional 3½-in. insulating fill 130

Floors over exposed or unheated spaces:

No. 15*a*. Double floor, on joists...................................... 42

b. Same as (15*a*) construction with additional ½-in. fibrous board fastened to bottom of joists....................................... 88

c. Same as (15*a*) construction with sheathing fastened to bottom of joists and with additional 3½-in. insulating fill between joists....140

The substitution of ½-in. insulating materials for sheathing should not be considered as having any additional value.

For walls and doors between heated and unheated spaces it is permissible to divide the value of *f* by 2.

Table 42. Air Changes To Be Assumed in Calculating the Value of *V*.

Description	Number of Air Changes per Hour
Living rooms with windows on one side	1
Living rooms with windows on two sides	1½
Living rooms with windows on three sides	2
Sleeping rooms	1
Entrance halls	2

Design of Wall Stacks. This is a simple matter because all first-floor stacks should have a free area equal to the basement pipe area and for all second- and third-floor stacks the free area should be equal to not less than 70 per cent of the basement pipe area. The framing of a structure sometimes makes it necessary to depart from these rules but they should be followed as closely as possible.

Design of Registers. A full description of registers and their design is given elsewhere in this book.

Design of Cold-Air Pipes. The total area of all cold-air pipes, when the return air is taken from the inside of the building, should be equal to the total area of all hot-air pipes. When air is taken from the outside, the cold-air pipe should be approximately equal to 80 per cent of the total area of all hot-air pipes.

Design of Gravity Hot-Air Furnace. The following formula is used in the Code method to design the furnace size. The constants in Formula **(27)** have all been determined by trial and experiment and give good results in ordinary conditions.

$$L=1.75G[1+0.02(R-20)] \qquad \textbf{(27)}$$

where

L=total area of warm-air pipes in square inches.

1.75=a constant based upon the results obtained in research on a furnace having 20 square feet of heating surface for each square foot of grate.

G=grate area of furnace in square inches.

R=ratio of heating surface to grate area.

The formula is based on efficiency of heater, 55 per cent; combustion rate, 7.5 pounds of coal per square foot of grate per hour; calorific value of fuel, 12,790 B.t.u. per pound; percentage of heat available at register, 75 per cent; average B.t.u. delivering value of one square inch of leader pipe area, assuming half of the heat is sent to each floor, 136; and on an operating temperature of 175°F. at the register.

The formula allows 1.75 square inches of warm-air pipe area for each square inch of grate area, for the furnace having a ratio of heating surface to grate surface of 20 to 1. For furnaces having other ratios of heating surface to grate surface, it adds 2 per cent or deducts 2 per cent for each unit above or below a ratio of 20.

Table 43. Explanation of Formula 27

	Positive Correction	No Correction	Negative Correction
Grate area, square inches	346	346	346
Heating surface area, square inches	7540	6920	5665
Ratio heating surface area to grate area	21.8 to 1	20.0 to 1	16.4 to 1
R-20	1.8	0.0	−3.6
Correction, per cent	3.6	0.0	−7.2
1.75 G	606	606	606
L=1.75G+Correction	628	606	562

Limitations of Code. Formula **(26)**, for determining the size of basement warm air pipes, is applicable to rooms of the proportions found in the average residence. For rooms having ratios of glass to cubic contents falling outside of these average proportions, adjustment must be made in the number of air changes to be used. The formulas are not applicable to pipes having diameters greater than 14 inches or lengths greater than 12 to 16 feet.

Formula **(27)**, the rating formula, is applicable to furnaces of the common types having ratios (heating surfaces to grate area) of between 15 and 30. The formula is not applicable to furnaces of special construction, to those equipped with unusual special features, to those having ratios outside of 15

to 30, nor to coals deviating materially from 12,000 B.t.u. per pound. In cases where any special conditions exist, the B.t.u. method of design should be used.

Location of Furnace. The location of a furnace, in a basement for example, should be such that the lengths of all warm air pipes will be as nearly equal as possible. Any inequality should benefit the rooms that are hard to heat. Fig. 46 shows a typical layout of furnace, leaders, and cold-air pipe. The furnace has been placed in a central location in the basement so as to make most of the leaders about the same length.

Installation of Furnaces. Furnace foundations should be of brick, concrete, or other like incombustible material, and should extend at least 15 inches at the rear and sides of the furnace casing and at least 36 inches in front of the furnace. Foundations should be perfectly level.

Whenever it is necessary to place a furnace on a combustible floor, a foundation of not less than 4-inch hollow tile should be used. Joints of the tile must be matched in such a way that air passage will be free from side to side, so that at no time will the removal of ashes or the handling of coal close up these openings. Such foundation must be constructed upon, and covered with continuous sheet-metal plates, of not less than No. 24 gauge metal, having all joints substantially riveted or double seamed, and the bottom sheet should have the edges turned up at least one inch. This floor covering should extend under the whole of the fire box and ash pit of the furnace and outwardly not less than 12 inches on all sides and rear of casting, or base ring, and 36 inches in front.

The base ring of any portable warm-air furnace should be cemented to the foundation, and cement flushed in around the back of the base ring, making an air-tight joint. The furnace parts should be assembled plumb and level and in a workmanlike manner. All sections and joints must be properly fitted. Joints requiring cement must be well filled and all bolts should be drawn up tightly. Warm air furnaces must be enclosed in metal casings or walls of brick, tile or concrete.

Sheet metal casings for portable warm air furnaces, including casing tops or bonnets, should be made of galvanized sheets, not lighter than 26-U.S. Standard Gauge. They must fit the castings and casing rings closely, so as to be dust tight, and must be securely fastened to the front. The casing must be lined from the upper casing ring down to a line on a level with the grate. When side collars are used, the casing top or bonnet must be of sufficient height so that the largest warm-air pipe can be taken from the side without ovaling. In no case should a distance less than eight inches (8″) be maintained between the top of any furnace and the bonnet.

Any furnace, the casing top of which shall come within twelve inches (12″) of a combustible floor, ceiling or joist, should be protected by a metal shield, extending not less than eighteen inches (18″) beyond the casing of said furnace. This shield should be suspended at least two inches below woodwork, allowing free air space between shield and woodwork. No furnace casing or top, coming nearer than six inches (6″) to ceiling or joists should be allowed in any case. All metal casing tops must be insulated with an air space, or covered with magnesia, asbestos boiler covering, or sand.

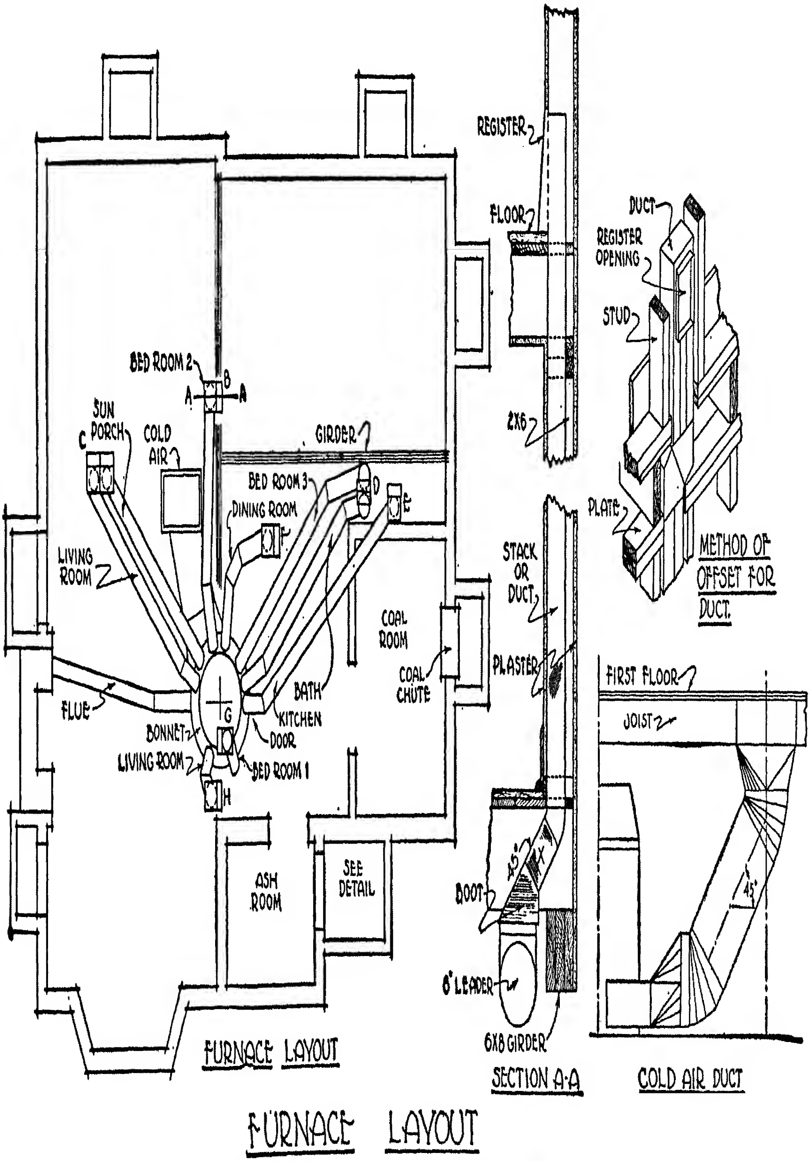


Fig. 46. Details of Furnace and Duct Layout for Cox Residence

Openings for side casing collars must be cut into the casing top or bonnet, so that the tops of all openings are on a level.

All warm-air pipes should be made of bright tin or galvanized iron. Side seams must be locked. All joints must be either double-seamed or lapped not less than one and one-quarter inches. All pipes should be properly secured to ceiling or joist. No solder or riveted joint is required where round pipe slips over the casing collar or enters boat or box. Any pipe 12 inches or greater in diameter should not be made of material lighter than IX tin or No. 26 U.S. Standard Gauge galvanized iron.

All warm-air pipes in the basement must have an upward pitch of not less than one inch per running foot.

No warm-air pipe should run within one inch of any woodwork unless such woodwork is covered with asbestos paper and the paper covered with tin or iron.

All warm-air pipes in the basement should be provided with dampers, supported on both sides, not more than two feet from the casing.

Where warm-air pipes pass through a masonry wall, a metal thimble should be provided, having a diameter at least one inch greater than the pipe, and the pipe should be supported in such a manner that the air space is uniform on all sides.

Where warm-air pipes pass through or into unheated spaces separated from the furnace room, they must be insulated with not less than three layers of air-cell asbestos paper or the equivalent.

All single stacks or wall pipes, heads, boots, ells, tees, angles and other connections must be made of tin or galvanized iron and must be covered with not less than one thickness of 12 pounds per one hundred (100) square feet of asbestos paper. All such stacks must be braced in a proper manner so as not to obstruct the flow of air but to retain the full capacity throughout. All joints should be locked and held in place by means of lugs, or straps. No joint, either horizontal or vertical, should depend wholly upon solder to make it tight. An air space of not less than five-sixteenths (5/16) of an inch must be allowed on the two sides nearest the vertical studs.

All double stacks or wall pipes, heads, boots, ells, tees, angles and other connections should be made of tin, not lighter than IC or galvanized iron and must be made double, from and including the boot or foot piece in basement to the top of each and every stack and register head on all floors. There should be uniform air space of not less than 5/16 of an inch, which must be maintained between the outer and inner walls of all such pipes and fittings of all kinds, styles and descriptions; such pipes, heads, boots and other fittings should be of the styles, or equal to those accepted by the National Board of Fire Underwriters.

All stacks and fittings, either single or double, must be secured firmly in place by lugs or straps attached to the outer walls of stacks and fittings, and no nails should be driven through these stacks or fittings at any point.

Where stacks, heads, boots, or other fittings go through the first floor, all openings around such must be filled with asbestos cement or other such incombustible material.

Registers must be properly sealed to the stack head in such a manner as to prevent any leakage of air betweeen the head and the register.

Registers should not be located in outside walls unless properly insulated with 1-inch air-cell covering or equivalent.

The air supply to the furnace may be taken from outside or from within the building or may be taken partially from outside and partially from within.

The cold-air intake or return where air is taken from within the building should have a net area throughout its entire length of not less than the combined net area of all warm-air pipes leading from the furnace. This may be maintained in one or more ducts. No reverse incline or air trap can be allowed in any section thereof.

When the cold-air supply is taken wholly from the outside of the building, the supply duct at its most contracted area must equal or exceed eighty per cent (80%) of the combined area of all warm-air pipes leading from the furnace.

Cold-air ducts should be constructed of metal, tile or other non-combustible material having a smooth inner surface, and must maintain a constant net area throughout their entire length. All joints must be made dust tight. Horizontal rectangular return ducts should have at least 10 per cent greater area than vertical connecting pipes.

Where a boot or shoe is connected to the casing at the base, the opening should not extend higher than a line on the level of the top of the grate of the furnace. The width of the shoe must be of proper measurement to make the area at all points at least equal to that of the round or square pipe to which it is connected. This boot or shoe should be of streamline transition construction.

Wherever the space between joists is used to convey cold air over head, all bridging and bracing should be removed and a sheet-metal pan should be constructed to extend not less than two inches (2″) below said joists. The connection from this pan to the boot or shoe should be made of galvanized iron not lighter than No. 26 U.S. Standard Gauge, and must have a transition fitting, the top area of which should be at least 10 per cent greater than the area of the connecting pipe.

Note: To reduce friction, and for the sake of cleanliness, it is recommended that the joists and all wooden surfaces between such joists be lined with metal.

When it is necessary to set the furnace over a pit and connect up cold air under the basement floor, such pit or cold-air trench shall not exceed eighteen inches (18″) in depth below the casing base ring, and the width of the trench or trenches should be of proper measurement to make the area at least 10 per cent greater than the pipe to which it is connected. The connection between the cold-air pipe or duct and the underground pit should be made with a transition fitting as described.

The cold-air face or faces should be made of wood, or metal. When set in floors, the top of same should be flush with floor. Where the cold-air face is placed in a seat or side wall (whether furnished by owner, general contractor or furnace contractor) the open work of the face must extend to within at least one inch (1″) of the floor line. The free area of cold-air faces

must be at least equal to the free area of the duct or ducts to which they are connected.

The effective area of a vertical cold-air face lies within fourteen inches (14") of the floor line, hence, the capacity of any vertical cold-air face should be determined by multiplying the base line in inches by not to exceed fourteen inches (14") in height and deducting for the grilles or cross bars.

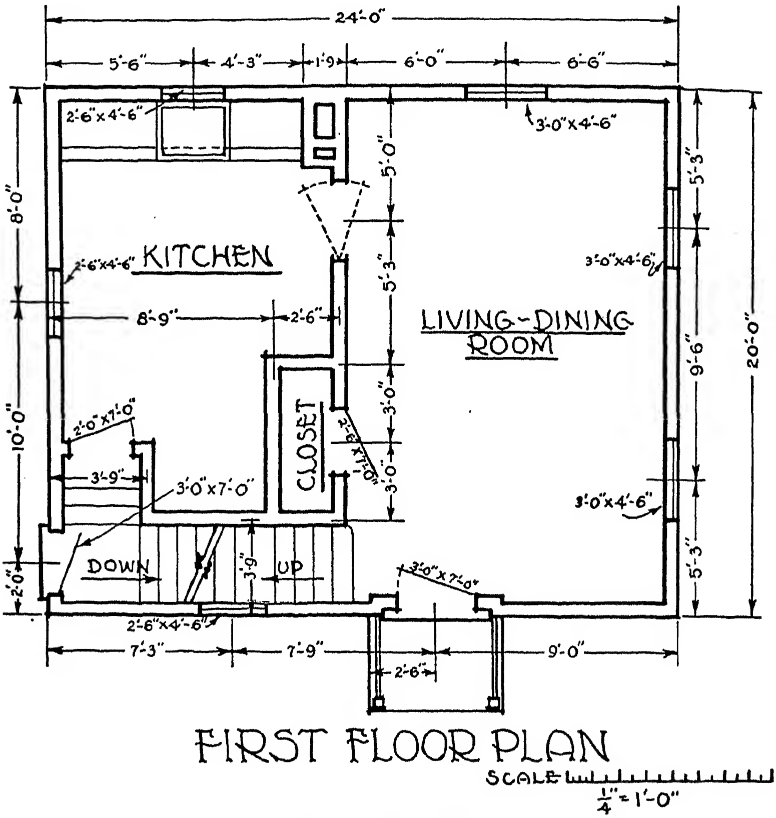


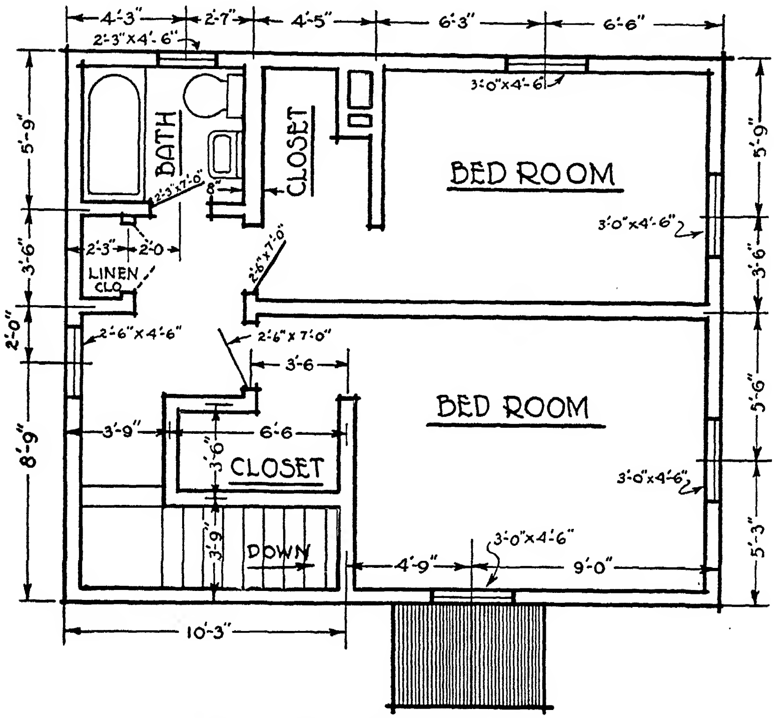
Fig. 47. First Floor Plan

The smoke pipe shall be as short and direct as consistent with the location of the furnace. It should be made of metal not lighter than No. 24 U.S. Standard Gauge, and not less than the full size of the collar on the furnace throughout its entire length. It must have no opening for attaching any fireplace, stove, range, water heater, gas or ventilating connection. It must be lock-seamed or riveted; all joints must lap not less than one and one-half inches (1½") and must be rigidly secured. Cast-iron smoke pipe may be used.

All smoke pipes should be provided with check dampers, placed on the side of the pipe or at the end of a tee; when cast-iron smoke pipe dampers are

used, they must be placed between the check damper and the furnace and supported on both sides of the pipe.

Where the smoke pipe enters the flue, a thimble should be cemented into the flue and the connections thereto made air tight. Should any smoke pipe come within eighteen inches (18″) of any combustible material, such combustible material must be covered with asbestos paper and a metal shield so



SECOND FLOOR PLAN

Fig. 48. Second Floor Plan

fastened that a two-inch air space exists between this shield and the combustible material.

"Humidification" and also "Design of Registers" are described completely in another section of this text.

Application of Design Methods to Common Types of Residences. The two methods (B.t.u. and Code) of designing leader sizes, furnace sizes, etc., which have been explained for gravity furnaces in the preceding pages, are illustrated by applying them to common types of residences.

The B.t.u. method is used in designing the heating system for Fig. 122. The Code method is used in the calculations for the residence illustrated by Figs. 47, 48, and 49, the specifications for which are as follows:

Walls: Shingles on sheathing on 2x4 studs, with ½-inch Insulite plaster backing and plaster.

Second-Floor Ceiling: ½-inch Insulite plaster backing and plaster. No floor above.

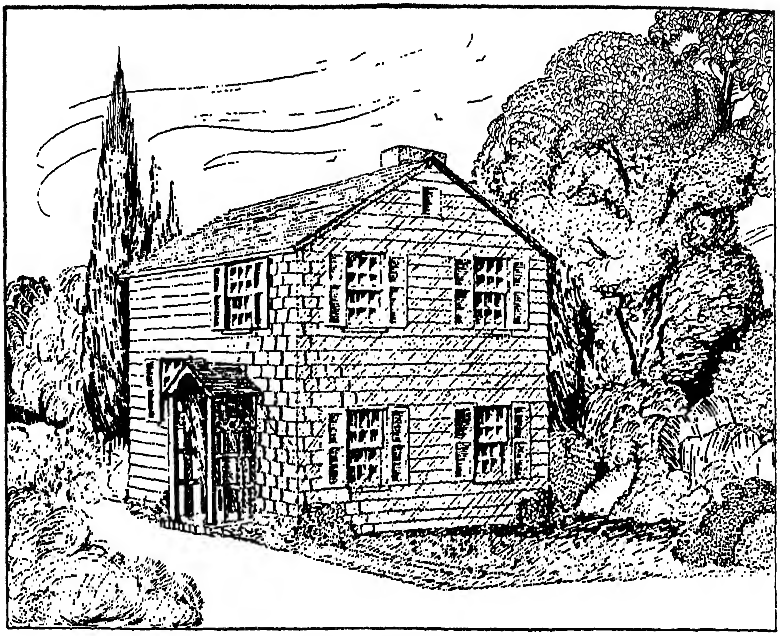


Fig. 49. Elevation

First Floor: Rough and finish flooring. ½-inch Insulite applied to bottom of joists.

Doors: Wood; actual thickness, 1⅝ inches.

Windows: Single glass, double-hung, no weatherstripping.

The basement may be considered 32°F., although it will probably be somewhat warmer because of heat loss from the furnace and leaders.

Chicago may be considered the geographical location.

The ceilings on both first and second floors are 10 feet high.

Register temperatures can be assumed as 175°F.

Baseboard registers, as shown in section on registers, are assumed.

The B.t.u. Method. The B.t.u. method requires that the heat losses, in terms of B.t.u. per hour, be calculated first. Examples of the calculation of heat losses are the factory building illustrative example in Chapter VI, Vol. II, and the example for Fig. 122. The two examples just specified illustrate the calculation of heat losses in all details except for structures having two or more stories.

In a residence, for example, having both a first and a second floor, the

first-floor rooms are handled exactly as for Fig. 122, except that no loss is assumed through the first-floor ceiling because the second-floor rooms above are warm. For the second-floor rooms, the process is the same as for Fig. 122, except that no loss is assumed through the floors because the first-floor rooms below are warm.

The following rules can be followed in calculating heat losses.

One story structures. Calculate losses due to

(1) Exposed walls

(2) Ceilings: Determine proper coefficient for pitched roofs by Formula (8), Vol. II.

(3) Floors: Determine basement temperature, or temperature under floor if there is no basement and calculate $(t\text{-}t_0)$ accordingly.

(4) Glass and doors (exposed)

(5) Infiltration

(6) For unusual exposure of any room or part of the structure, 15 per cent is added to the heat loss for that room or part.

Two story structures. Calculate losses due to

(First Floor)

(1) Exposed walls
(2) Floors: Determined as for one-story structures.
(3) Glass and doors (exposed)
(4) Infiltration
(5) Unusual exposure

(Second Floor)

(1) Exposed walls
(2) Ceilings: Determined as for one-story structure.
(3) Glass and doors (exposed)
(4) Infiltration
(5) Unusual exposure

Consideration must sometimes be given to such items as the following: Doors leading to attics and basements will have a different $(t\text{-}t_0)$ value than exposed or inside doors. Ceilings of closets should be included with the ceiling area of the adjacent room if the closets are large. The same applies for closet floors. Thus in Fig. 48 ceilings of the closets for the two bedrooms must be included with the adjacent bedrooms because the closets are large. The exposed wall of the closet for the rear bedroom must be included in the exposed wall area for the rear bedroom. Small closets do not require such attention. Stair wells, in structures having two or more stories such as Figs. 47 to 49, should be considered, especially where the stair well has an exposed wall and a window as part of it. The amount of wall surface above the treads and risers, plus loss through glass and by infiltration, might well be added to the loss for the living room so that ample leader size will be supplied to warm the stair well ana the upstairs hall. Special consideration is necessary for each individual structure to determine what to include in heat-loss calculations. In the case of the upstairs hall in Fig. 48, for example, no hot-air register is deemed necessary.

After the heat losses have been determined for each room of a structure, the leader, cold-air return pipe, stack, register, and furnace sizes are determined as explained on pages 59 to 65.

The placing of stacks, returns, and the furnace, is explained in the following example where the Code method is employed.

As mentioned previously, the B.t.u. method is an accurate but painstaking and laborious method for designing gravity warm-air systems, and need only be employed where very special or unusual conditions are met, which cannot be considered in the Code method.

The Code Method. As already explained the Code method is a simplified design procedure. It saves much labor and time in calculating a gravity warm-air system. It can be employed successfully for all ordinary structures where no special conditions exist and where the contemplated system falls within its specifications as, for example, 175°F. register temperatures. Because of the simplicity of gravity systems, the Code method is applicable in most cases. The following example illustrates its use.

Example. Use the Code method to design the leaders, stacks, cold-air returns, registers, and furnace size, using a Moncrief Furnace*, as shown in Fig. 45, for the residence shown in Figs. 47, 48, and 49. Show locations of all registers, stacks, cold-air returns, and the furnace. The following data applies to this example.

(1) Location——————————————Chicago

(2) Lowest outside temperature—23°F. (Chapter VI, Vol. II, Table 25).

(3) Base temperature: In *this example* a design temperature 15°F. above lowest on record is used. Hence base temperature$=(-23+15)=-8$°F.

(4) Inside air: The temperature of 70°F. is used in all cases instead of figuring exact temperatures at various levels.

(5) Dimensions: See Figs. 47 and 48.

(6) Construction: As noted in specifications. Walls: Shingles, sheathing, 2x4's, ½-inch Insulite plaster backing and plaster.

First Floor: Rough and finish flooring. ½-inch Insulite applied to under side of joists.

Second Floor Ceiling: ½-inch Insulite plaster backing and plaster. No floor above.

Doors: Wood; 1⅝ inches thick.

Windows: Single glass, double-hung. No weatherstripping.

All ceilings are 10 feet high.

Solution. The first step in the solution of the example is that of calculating the various areas required in Formula (26). Table 44 has been calculated as follows. Figures in Table 44 are approximately correct. The living room is calculated first. When calculating the wall area for this room, it is necessary to include the wall area taken by the stair well on the front elevation. This is an exposed wall with a window.

The living room (inside dimensions) is 12 feet wide by 19 feet long. The area of one end is 12x10=120 square feet. The area of the long side is

* Data Courtesy of The Henry Furnace and Foundry Co. Cleveland.

19x10=190 square feet. There are two exposed ends and one long side, or 120+120+190=430 square feet. The wall (inside dimensions) taken by the stairway is 1'9"+4'3"+5'0"=11 feet long. Then 11x10=110 square feet. The stairs slant upward so that they divide the wall diagonally in half.

*** Table 44. Calculation Sheet for Figs. 47 and 48 Code Method**

ROOM	1 Floor Area Square Feet	2 Ceiling Area Square Feet	3 Wall Area (Net) Square Feet	4 Door Area Square Feet	5 Glass Area Square Feet	6 Volume Cubic Feet
Living room	228	...	414[1]	21	50	2465[2]
Kitchen	136	...	249	14	22	1360
Bedroom (front)	...	157	204	..	26	1570
Bedroom (rear)	...	135	209	..	26	1350
Bath	...	31	100	..	10	310

*Figures are approximate.
[1] Area includes exposed wall area in stair well.
[2] Volume includes partial stair-well volume. Thus half the area becomes basement area and half living room area. It is considered feasible to assume that only half of 110 square feet need be added to the living room wall area. Therefore 55 square feet are added to 430, making 485 square feet. This 485 square feet includes door and windows. To find *net* wall area the area of door and windows must be subtracted from 485.

In the living room there are three windows, each 3'0"x4'6". The area of one equals 3'0"x4'6"=approximately 13 square feet. Three windows have a combined area of 3x13=39 square feet. The window in the stairway has an area of 2'6"x4'6"=11 square feet. Total window area equals 39+11=50 square feet. This area is put in Column 5 of Table 44.

The area of the door is

3x7=21 square feet

This is put in Column 4 of Table 44.

Combined window and door area=50+21=71 square feet. Then the net wall area is

485—71=414 square feet

This is put in Column 3 of Table 44.

The floor area of the living room is

12x19=228 square feet

This is put in Column 1 of Table 44.

By this time it has become apparent to the reader that only walls, windows, doors, etc., that are exposed to outside temperatures or to temperatures below 70°F. are considered.

The volume of the living room is calculated to include the room itself plus the stair well. Because the stairs slant upward, cutting the stair-well area about in half, only one-half of the stair-well volume is used. The room volume is 19 feet (length)x12 feet (width)x10 feet (height)=2280 cubic feet. The stair well is 11'3"(length)x3'3"(width)x10'0"(height)=370 cubic feet (approximately). One-half of 370 equals 185 cubic feet. The entire volume for living room is then

2280+185=2465 cubic feet

The living room is the only room having additional volume added to it. Volumes of all other rooms are calculated by multiplying length by width by height.

The kitchen is irregular in shape and has one side facing a stair well. This is an unusual condition requiring a special explanation.

Part of the end of the kitchen facing the stair well is exposed to a 32°F. temperature; part is exposed to 70°F. This is because the stairs (which are warmed) going to the second floor are directly above the basement stairs and run parallel to them. A cross-sectional sketch taken through the long dimension of the stair well would show this if the reader has trouble in visualizing the situation.

In this example, unlike the example for Fig. 122 (where outside temperature was assumed for the basement stairs), it is assumed that the wall adjacent to the stair well comes in contact with a 32°F. temperature instead of outdoor conditions. For this reason, special consideration is required.

The two walls of the kitchen which contain windows, are exposed to outdoor conditions and thus are subject to the 78° temperature difference. The wall next to the basement stair well is exposed to a temperature difference of only 38°. Thus, if no special condition existed, it would be necessary to calculate separately the heat losses through the walls exposed to the outdoor conditions and the wall exposed to the stair well. However, as part of the wall exposed to the stair well comes in contact with air at 70°F., a special condition exists which can be safely handled by estimating, as explained in the following.

By scaling and by dimensions, the left-hand wall between the kitchen door (to stairs) and the corner by the sink cupboards, is 13′0″ long. The sink wall is 9′3″ long. Then 13′0″+9′3″ equals 22′3″, total length of kitchen wall exposed to outdoor conditions. For the wall exposed to the stair well, it is safe to assume a certain amount of it (that containing the door and that at right angles to the door) as if exposed to outdoor conditions instead of attempting to calculate the exact amount exposed to the 32°F. condition. That part of this wall containing the door, plus the short wall at right angles to the door wall, scale 3′6″ and 2′9″ or a total of 6′3″. Adding this length of wall to 22′3″ gives 28′6″. Then 28′6″×10′0″=285 square feet gross area.

Note: The foregoing method of determining how much of the wall adjacent to the stair well should be included with the wall exposed to outdoor conditions, is purely an estimate. There is no set rule to follow in such cases, therefore the engineer must use his own judgment. In this case the estimate is surely on the safe side, especially when it is remembered that the 32°F. assumed basement and stair-well temperature is a large factor of safety due to the fact that the temperature of these areas no doubt will be considerably above 32°F. most of the time.

To find the net wall area, the door (leading to stairs) and window areas must be subtracted from the gross area. The net wall area is

$$285-36=249 \text{ square feet}$$

This is put in Column 3 of Table 44.

In like manner, areas and volumes of all other rooms are calculated and the results are entered in Table 44.

Having calculated all necessary areas and volumes, the next step is the calculation of leader sizes by Formula **(26)**. This formula is used to design leader sizes without the necessity of calculating heat losses. The explanation of the letters in Formula **(26)** are given on p. 68, so they are not repeated here.

Formula **(26)**

$$A=\left(\frac{G}{12}+\frac{W}{f}+\frac{C}{f}+\frac{F}{f}+\frac{V}{800}\right)\times K$$

The living room is considered first.
Substituting in the formula:

$$A=\left(\frac{71}{12}+\frac{414}{80}+\frac{228}{176}+\frac{3698}{800}\right)\times 9$$

The glass plus door area (G) is 71 square feet. (See Table 44.)

The glass area, 50 square feet, and the door area, 21 square feet, are added together to give 71 square feet. The 71 is put in the formula in place of G.

The net outside wall area is 414 square feet. This is put in the formula in place of W.

The ceiling for this first-floor room is below the heated second-floor rooms so the quantity $\frac{C}{f}$ is not considered.

The floor area is 228 square feet. This is put in the formula in place of F.

The volume of the living room is 2465 cubic feet and it has windows on two sides. (See Table 42.) Then $2465\times 1\frac{1}{2}=3698$ cubic feet. The 3698 is put in the formula in place of V.

The value of f for outside walls is 80. (See Table 41.) The 80 is put under the 414.

Table 41 shows the value of f for the floor is 88. This is doubled because the basement is not as cold as outside.

The value of K is 9 because the living room is a first-floor room.

Substitution in the formula is now complete and it is solved as follows—

$$A=\left(\frac{71}{12}+\frac{414}{80}+\frac{228}{176}+\frac{3698}{800}\right)\times 9$$

$$A=(5.92+5.18+1.30+4.62)\times 9$$

$$A=(17.02)\times 9$$

$$A=153.18$$

Note: Note 2, p. 68 specifies that some consideration must be given where the temperature difference is greater than 70 degrees. In this example the difference is 78 degrees. Therefore $8\times .015=.12$ and $.12\times 153.18=18.38$. This is added to 153.18 making it 171.56. Call it 172 square inches.

The areas of the leader pipes for all other rooms are calculated as explained for the living room, keeping in mind that the value of K is 6 for second-floor rooms.

The leader areas for the various rooms are shown in Table 45.

Table 45. Leader Pipe Areas

ROOM	Leader Pipe Square Inches
*Living room	172†
*Kitchen	95†
Bedroom (front)	60†
Bedroom (rear)	56†
Bath	20†

* Assume windows on two sides. See Table 42.
† Figures approximately correct. General practice makes for approximate figures where it is felt there is ample factor of safety.

It should be remembered that Note 3, p. 68 applies to all rooms. No unusual exposures are assumed.

The leader sizes for the various rooms are as shown in Table 46. They are based on Table 40. The smallest leader recommended is 8 inches. Note that special conditions must be provided for in the living room. Thus the living room leader would have to be 15 inches in diameter.

Table 46. Leader Pipes Required

ROOM	Number of Leaders	Diameter of Leaders Inches
Living room	2	11*
Kitchen	1	11
Bedroom (front)	1	9
Bedroom (rear)	1	9
Bath	1	8

* A round leader pipe of 172 square inches (cross-sectional area) has a diameter of approximately 15 inches. See Table 40.

Moreover, even though the register were centrally located in the room, uneven heating would result because of such a large volume of heated air being delivered at one location. Also, unless the register were located near the stairs, the stair well would not receive sufficient heat to make the hall and stair well comfortable.

The recommended procedure, where a leader of over 12 inches is necessary in a large or long room, is to use two leaders, each of which delivers half the required amount of heated air. This plan assures even heating of the room, makes large space-taking leaders unnecessary, and at the same time, in this example, insures an adequate supply of heat for the stair well.

Thus ½ of 172=86. From Table 40 the nearest pipe size is one having a diameter of 11 inches. So, two leaders, each having a diameter of 11 inches are selected.

Table 40 shows that the leader pipes chosen have areas as shown in Table 47.

It should be remembered that the leaders selected are slightly more in area than required. This is so because the nearest larger size of standard

pipe was selected, and, in the case of the bathroom, because the smallest recommended pipe had to be specified, even though it is much too large. The sum of the selected leaders (463 square inches) is used in designing the furnace size.

The stack sizes are designed next. The Code says that stacks going to the second floor should be not less than 70 per cent of the basement leader pipe area.

Table 47. Areas of Required Leader Pipes

ROOM	Number of Leaders	Diameter of Leaders Inches	Areas in Square Inches
Living room	2	11*	190
Kitchen	1	11*	95
Bedroom (front)	1	9	64
Bedroom (rear)	1	9	64
Bath	1	8	50

*Commercial sizes for leaders are 5, 6, 7, 8, 9, 10, 12, 14, 16, etc., in multiples of 2 starting from 10. In this problem the living room and kitchen leaders are not commercial sizes. If commercial sizes are required, the next largest size, or 12-inch leaders, would be selected. This would add considerably to the capacity of the leaders and increase the factor of safety. In this problem the 11-inch leaders are employed and the furnace size based on their use. It should be remembered that an increase in leader sizes also increases the furnace size.

The bedrooms have leaders 64 square inches in area. The 2x4 partition studs are spaced 16 inches center to center. If we allow a clear space of 14 inches between studs and a width of 3½ inches, the area of the stack is $14 \times 3\frac{1}{2} = 49$ square inches. This is amply large because 70 per cent of 64 square inches is approximately 45 square inches. Therefore the stacks can be $14 \times 3\frac{1}{2}$ inches in cross section. Slight variations of these dimensions are possible according to actual clear area between studs where the stacks must be installed.

The Code specifies that cold-air returns must have the same aggregate area as the leaders. In other words, the sum of the areas of all cold-air returns should be equal to the sum of the areas of all leaders. This rule must be followed in systems where the cold-air pipes are joined to the furnace as shown in Fig. 40. In this example, the cold-air pipes end at a point 18 inches above the basement floor. Therefore, much less friction needs to be overcome than when the cold-air pipes are joined to the furnace. Fig. 50 is a diagram showing the circulation of air in a system where the returns are left open in the basement. Actual trials have proved that if the cold-air returns total two-thirds the area of the leaders, the system will function properly.

In a system such as illustrated by this problem, the basement must be fairly open and not divided into various rooms. Where basement rooms are to be built, the cold-air pipes must go directly to the furnace casing.

A furnace requires a constant supply of air equal to the amount of air required by the structure it is heating. Thus if a residence has a volume of 20,000 cubic feet (sum of volumes for all rooms) and there are 1½ changes of air per hour, the furnace will require 30,000 cubic feet of air per hour. This air can be supplied by recirculating the air in the residence, by taking the

air from the outside, or by recirculating half the required air and taking the other half from the outside. When all the air is taken from the outside, the heat loss (total) for the residence should be increased by 25 per cent. A discussion of "Recirculation of Air" can be found under that heading in Chapter IV, p. 48.

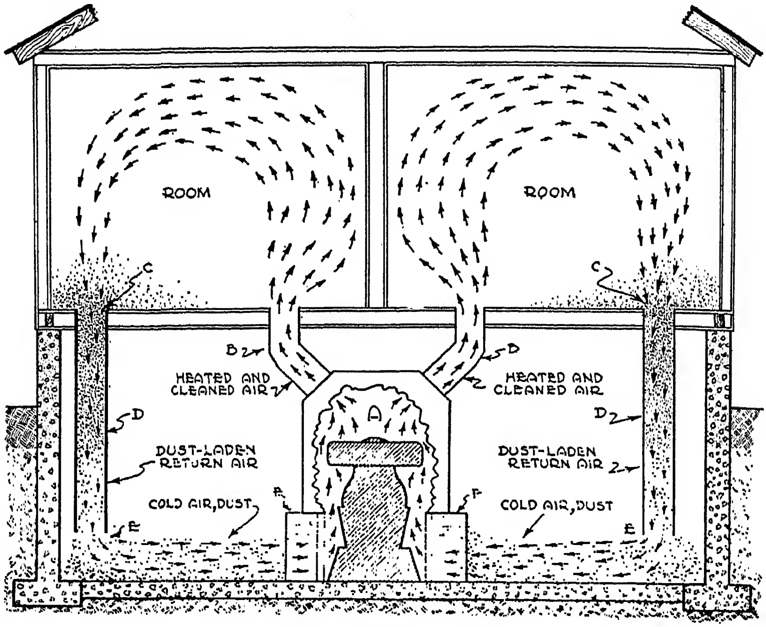


Fig. 50. Circulation of Air

In this problem the air is recirculated. The cold-air pipes (drops) end 18 inches above the floor. See Fig. 52. Other systems employ the method whereby one or more cold-air pipes are joined to the furnace casing. In compact structures, one large cold-air pipe is sometimes used, but in structures where the rooms are widespread, more than one cold-air pipe is required. In all cases it is best to put the cold-air pipes in outside walls and the cold-air faces (registers) in the outside walls, where they are near the greatest area of cold air.

In this problem, it was decided to have cold-air faces for the living room and the two bedrooms. See Figs. 51, 52, and 53. This decision was reached after noting that the kitchen and bathroom air cannot be recirculated because of objectionable odors. The hall on the second floor is not considered because of its small size. Thus only the living room and the two bedrooms remain. As a trial, based on judgment and experience, the cold-air returns are designed as shown in Figs. 51, 52, and 53.

Note: Many trials may be necessary before a suitable design is found. The design shown in this problem is assumed to have been selected after one or more trials.

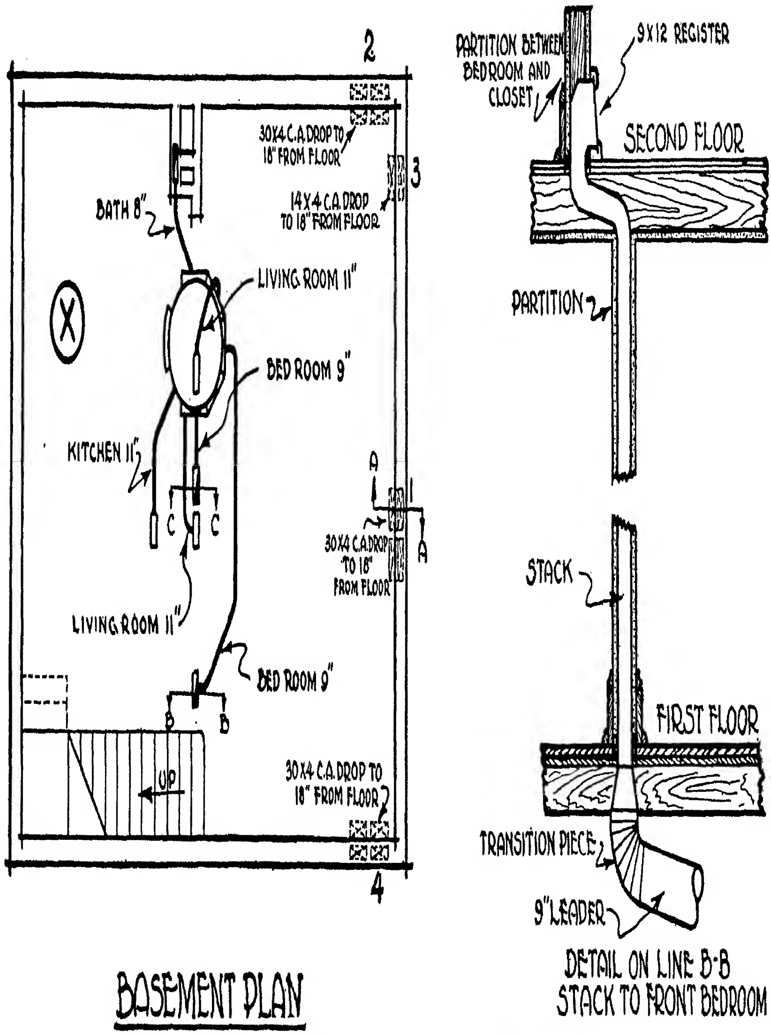


Fig. 51. Plan and Details of Cold and Warm Air Pipes in Basement

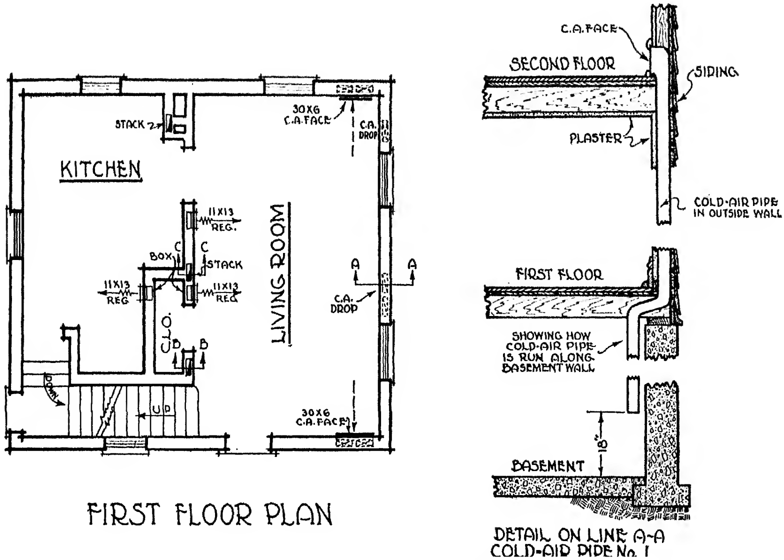


Fig. 52. Arrangement of Cold and Warm Air Outlets and Inlets and Details for First Floor

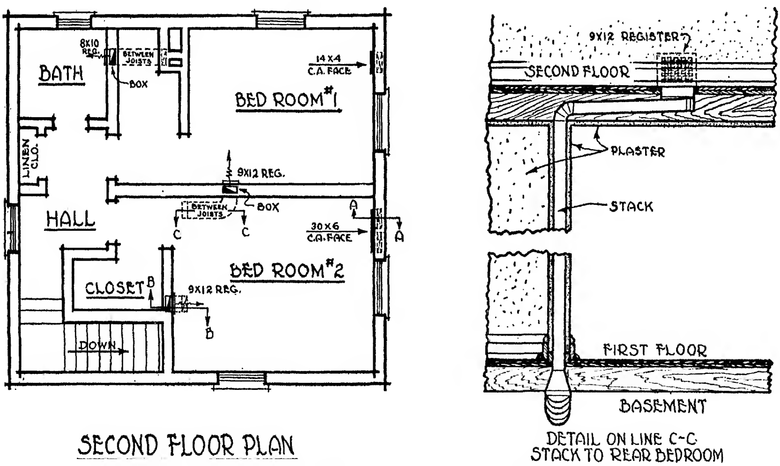


Fig. 53. Arrangement of Cold and Warm Air Outlets and Inlets and Details for Second Floor

The sum of the areas of the cold-air returns is as follows:

Drop 1 is $30\times4=120$ square inches
Drop 2 is $30\times4=120$ square inches
Drop 3 is $14\times4=\ 56$ square inches
Drop 4 is $30\times4=120$ square inches
Total area$=416$ square inches

This is considerably more than two-thirds of the total leader-pipe area (463 square inches) so it is sufficient and allows an ample factor of safety.

In Figs. 51, 52, and 53, drops, Nos. 1, 2, and 4 are each shown in two rectangles or parts. This is because the 2×4 studs are only 15 inches apart, thus requiring two seperate drops to make 120 square inches. In other words, each drop is $4\times15=60$ square inches. The two are joined at the face.

The question might arise as to why the cold-air returns should be placed as shown in Figs. 51, 52, and 53. The living room is long, so two sets of drops seemed advisable. Bedroom No. 2, being larger than bedroom No. 1, contains the larger drop.

The registers and cold-air faces must have a free area at least equal to the leaders supplying them. For the living room there are two leaders, each 95 square inches in area. From Table 69 in Chapter VIII, it is seen that register sizes of 11×13 inches are required. The kitchen leader is also 95 square inches in area, so it requires an 11×13-inch register. Each bedroom has a leader of 64 square inches, so each requires a register 9×12 inches. The bathroom leader is 50 square inches so its register must be 8×10 inches.

The living room has two cold-air pipes, each of which has an area of 120 square inches. From Table 70, it can be seen that a cold-air face 6×30 inches is required for each. In like manner the front bedroom requires a 6×30-inch cold-air face. The rear bedroom with a cold-air pipe area of 56 square inches needs a 14×4-inch cold-air face. This latter size is not shown in Table 70 but is listed in manufacturers' catalogues.

The furnace size is computed as follows—

Assuming a furnace rated at 20:1 and the same specifications as given under the explanation of Formula **(27)** the design of the furnace is a simple task.

The formula when using a furnace rated at 20:1 (no correction necessary) reduces to:

$$L=1.75G \tag{28}$$

The sum of the leader pipes selected for use, is 463 square inches. Substituting:

$$463=1.75G$$

$G=463\div1.75$

$G=265$ square inches, area of required grate.

A circle containing 265 square inches has a diameter of 18.4 (approximately).

Thus the required furnace has a grate diameter of 18.4 inches. From Table 38, a grate diameter of 20 inches is selected as the nearest safe standard

size. This would require a 24-inch furnace, as furnaces are rated on the largest diameter of firepot, which usually is 4 inches larger than the grate diameter.

The location of the hot-air registers should always be on inside walls if possible. If they must be put on outside walls, the boxes and stacks should be well insulated. The living room was given two leaders so as to deliver warm air more evenly. The delivery location was chosen on the inside walls so the hot air would flow over the room on its way to the cold-air faces. This causes a circulation which is required for good heating. In like manner all other registers were placed. Structural conditions also govern, to some

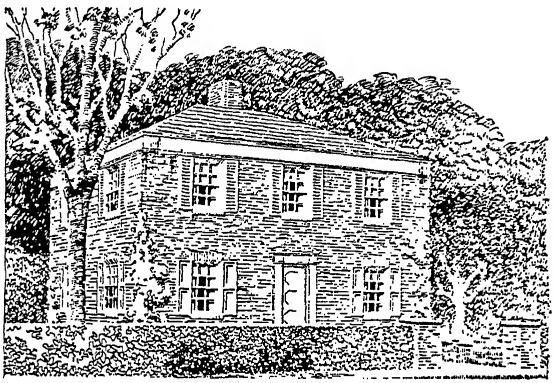


Fig. 54. Elevation

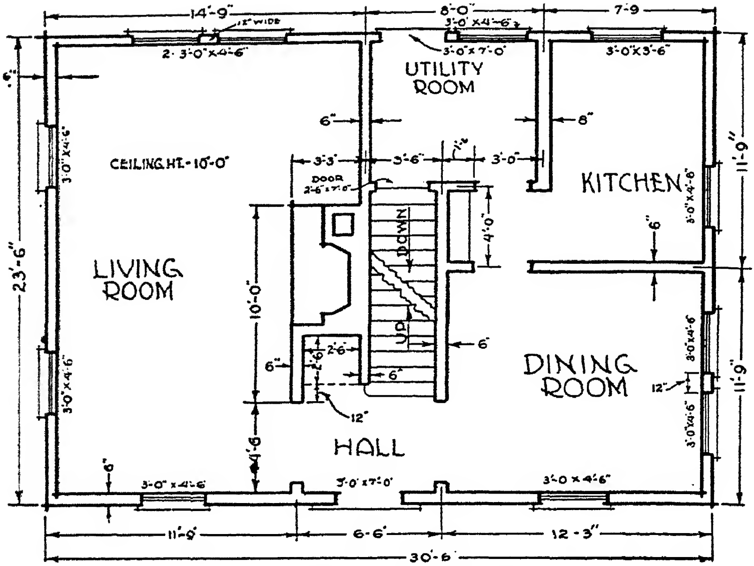
extent, the positions of registers, because the stacks must come up through partitions. The detail drawings in Figs. 51, 52, and 53 illustrate the procedure when partitions are not one over the other. Structural plans must be carefully studied so that stacks and leaders may be as short as possible.

The cold-air faces were put in outside walls opposite the registers for reasons already explained.

The furnace was centrally located, following Code specifications, so as to keep down the lengths of leaders. On the other hand, the furnace was located as far to the left-hand side of the basement as possible (without increasing length of leaders) to leave free space in the basement for the laundry, etc., and put the furnace near the fuel bin, located at X in Fig. 51. Locating the furnace requires careful study and the making of one or more rough sketches so that leader lengths can be accurately scaled. In this example the sides of the furnace containing the air filters were placed so that the filters faced the direction of the cold-air drops.

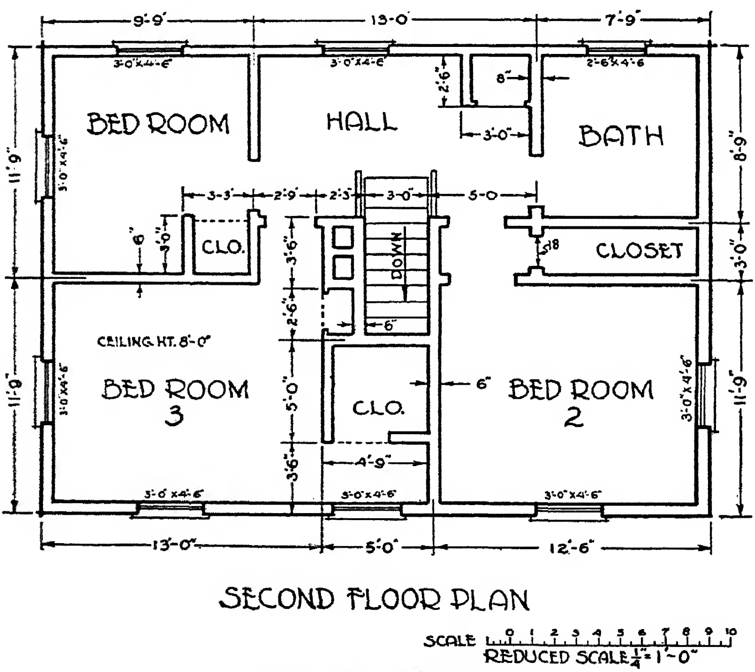
PRACTICE PROBLEMS

1. This problem consists of designing a gravity warm-air heating system for the residence shown in Figs. 54, 55, and 56. The furnace is to be a Moncrief Clean-Air Type with filters (Fig. 45). The cold-air drops end 18 inches above the floor.



FIRST FLOOR PLAN

Fig. 55. First Floor Plan



SECOND FLOOR PLAN

Fig. 56. Second Floor Plan

Use the Code method in calculating, designing, or selecting the following items.

(a) Leader sizes for every room
(b) Stack sizes
(c) Register sizes
(d) Cold-air returns
(e) Cold-air faces
(f) Furnace size (Furnace ratio is 22:1)

After items (a) to (f) have been decided upon, draw sketches (in pencil to the ¼″=1′0″ scale) of the basement, first-floor and second-floor plans, showing the following items:

(g) Location of all stacks
(h) Location of furnace
(i) Location of all leaders
(j) Location of all cold-air drops (Multiple system)
(k) Location of all registers and cold-air faces

Items g to k should be drawn like Figs. 51, 52, and 53 with all leaders, stacks, registers, faces, etc., named and the sizes given.

Note: When drawing the basement plan, show the chimney and stairs. No other structural parts, such as columns, need be shown.

Draw details (scale ½″=1′0″) showing the following:

(l) Cross-sectional drawing showing stack for Bedroom No. 1, from first floor level to the register.

(m) Cross-sectional drawing showing cold-air drop from point 18 inches above basement floor to its second-floor cold-air face.

Specifications.

Doors: All outside doors are 1⅝ inches thick and of wood.

Windows: All windows single glass and double-hung. Assume 1⁄16-inch crack and 3⁄64-inch clearance. No weatherstripping.

Walls: Western Frame. On the outside, sheathing, paper, and siding; on the inside, ½-inch Insulite plaster backing and plaster.

First Floor: ⅜-inch oak flooring on ⅞-inch yellow pine—sub-flooring on joists. ½-inch Insulite rigid insulation is nailed to bottom of joists.

Second Floor: Same flooring as for first floor, with wood, lath, and plaster.

Second Floor Ceiling: ½-inch Insulite plaster backing and plaster. There is a tight floor of ⅞-inch yellow pine above.

Roof: ⅝-inch yellow pine roof boards, one against the other, and asbestos shingles.

Basement: Unheated except what heat escapes from furnace and leaders. Assume a temperature of 32°F.

Attic: Assume temperature same as outdoors.

Location: Chicago, Illinois.

Lowest Temperature: —23°F.

Inside Temperature: 70°F. Use this temperature entirely, with no special calculations for special points. Assume no special exposure.

2. What special consideration must be given warm-air stacks that for structural reasons must be run in outside walls?

3. What must be done if leader must be longer than 12 feet?

4. Is it economical to use, say, about 75 per cent outside air in place of recirculating the air 100 per cent? Give discussion to prove your answer.

5. Suppose a second-floor room required a leader area of such size that a large enough stack could not be carried between studs in the partitions. How could this problem be handled?

Summary. In the problem illustrating the Code method, the filter system was used to clean the air. If the old-style return system had been used, the cold-air pipe or pipes would have been returned to the furnace. Fig. 46 showing details of furnace and duct layouts, illustrates the system of returning the air directly to the furnace. There is one large cold-air face, as noted. The detail at the lower right-hand side of Fig. 46 shows the cold-air pipe. In this case the cold-air pipe must have an area equal to the total area of all leaders. For a residence of larger size or where the floor plan is less compact, it would be wise to use two or more cold-air returns having a total area equal to the total area of all leaders. Then the second-floor returns could run down through the outside walls and the first-floor returns could be placed anywhere.

The number of cold-air returns to use in any given residence depends on the arrangement and number of the rooms. The cold air must easily reach cold-air returns from every room. Thus in a complicated floor plan of three or four rooms, or more, to each floor, it may be necessary to have a cold-air return for each room, unless one or two returns can be placed so that the cold air can easily find its way from all rooms to the returns. A room will not heat properly unless the cold air can easily find its way to a return.

Figs. 47 and 48 illustrate a small, compact house that could have one large cold-air return if the system shown in the foregoing problem were not used. With one cold-air return, the cold-air face must be in such a location as would allow the cold air from both floors to reach it. Such a location would be at the foot of the stairs. At that point, air from the first and second floors could reach it. The one-pipe cold-air return system, however, depends on room doors being open for easy flow of air, and for this reason the multiple (more than one) cold-air return system seems better in most cases.

Results obtained by the B.t.u. and Code methods cannot be com-

pared unless the conditions of the Code on such points as in filtration approximate the B.t.u. calculations for the same items. The B.t.u. method of calculating infiltration through door and window cracks does not always give the same results as the Code method of dividing the air change by 800. Also, the B.t.u. method uses more exact figures. However, the two methods will generally give results that are reasonably close.

In calculating heat losses, leader areas, furnace sizes, etc., using either the B.t.u. or Code method, there is much reasoning necessary as to what to include or consider in addition to outside walls, windows, floors, etc. In the case of Figs. 47 and 48, there is a stair well along an outside wall and an upstairs hall. The question arises as to what to do about these areas. In the Code example, consideration was given the stair well. In other words, larger leaders than actually required for the living room were selected, assuming that some of the excess warm air would make up for the heat loss through the stair-well wall and window and that a goodly part of it would find its way to the upstairs hall. It was assumed that warm air from the other upstairs rooms would also help to keep the hall warm. In the examples, the leader areas throughout are in excess of what is actually needed but the excess is a good safety factor for extremely cold or windy periods. Such a safety factor also provides for quick warming, when necessary, and allows a normal rather than forced operation of the furnace.

The B.t.u. method is perhaps better fitted for use in extreme or unusual construction or shapes of houses. Moreover where extreme care must be exercised the B.t.u. method can best be used unless the situation is such that the Code method fits in without any loss or gain because of conditions outside of the Code considerations.

Gravity warm-air heating is best suited to compact houses where no long leader runs are necessary. In houses having wings or ells, the problem of heating the rooms by gravity becomes impossible of solution. Long leaders, if used, must be of ample size, well pitched, and well insulated. Houses having exposed wings for example, may require leaders much longer than 12 feet (maximum recommended length); infiltration may create severe back drafts in the wings; and the basement ceiling height may not be sufficient to allow leaders to have a pitch of more than one inch per foot. Such

conditions really call for the use of other types of heating plants which are discussed in the following paragraphs.

Auxiliary Fan Systems. Auxiliary fans can be added to existing gravity furnaces by inserting the fan in one of the cold-air return pipes at a point just before the pipe enters the furnace casing. See Fig. 57. This system helps materially to heat rooms that might

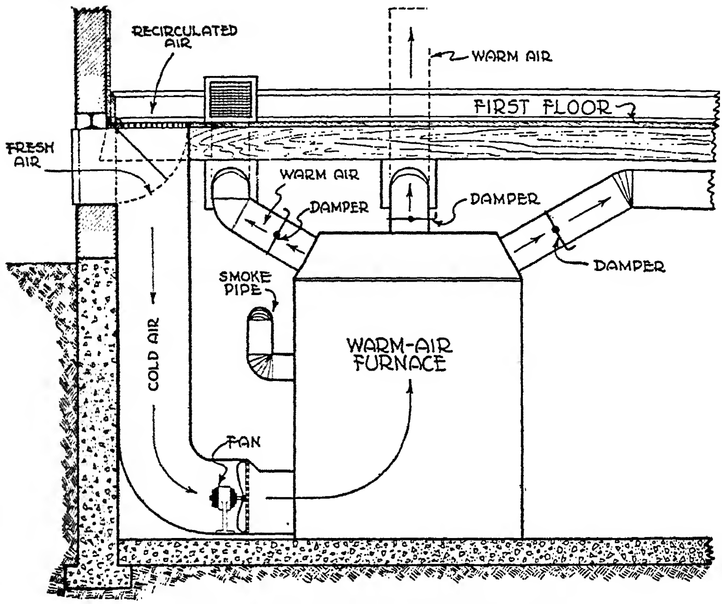
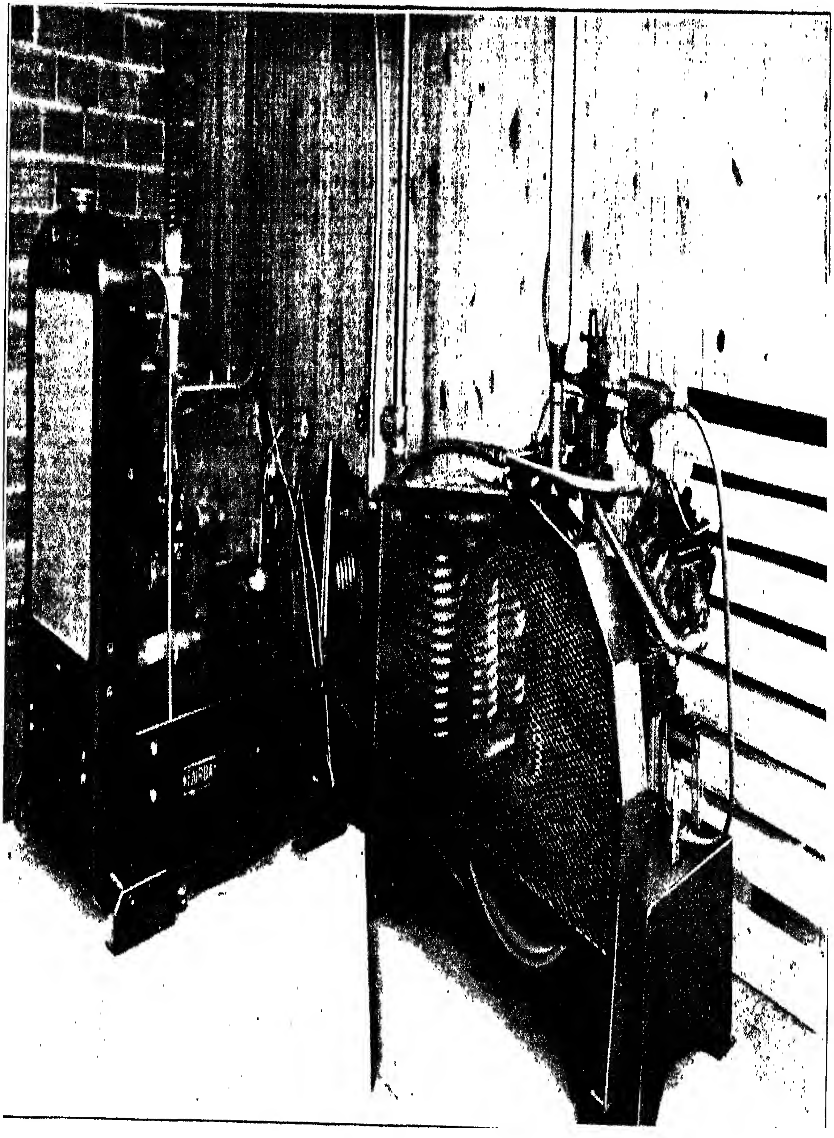


Fig. 57. Auxiliary Fan Installed in Old Gravity Furnace

otherwise be lacking in heat because of excessive leader length or because of wind direction, etc. However, rooms nearest to the furnace or having the least frictional resistance in the leaders will continue to receive more heat than the room farther away or having considerable frictional resistance in its leader. Thus the hardest room to heat will actually receive more heat than before, although still proportionately less. It can easily be seen that the rooms which are easy to heat will receive more heat than needed. Thus a condition would result which is no better than existed before the installation of the fan unless some additional means are provided to more evenly distribute the heated air. The remedy is to adjust the dampers, as shown in Fig. 57, so that the rooms easy to heat receive

less heat per hour. This will result in the easy-to-heat and hard-to-heat rooms getting their proper amount of heat at all times.

The degree of success possible with such a fan depends on other things, such as size of leaders, for example. If the leaders are much too small or have too many turns, or are much too long, it is doubtful if the fan would accomplish much good. Also, if the natural gravity flow of air is high, the fan might not improve conditions. Therefore the use of an auxiliary fan may or may not improve existing conditions. Before installation, a very careful study of conditions should be carried on.



A TYPICAL COMPRESSOR FOR CENTRAL PLANT

Courtesy of Fairbanks-Morse Company

CHAPTER VI

MECHANICAL WARM-AIR FURNACES

This type of furnace is similar to the ordinary (gravity) type furnace except that the air, instead of circulating by natural gravity,

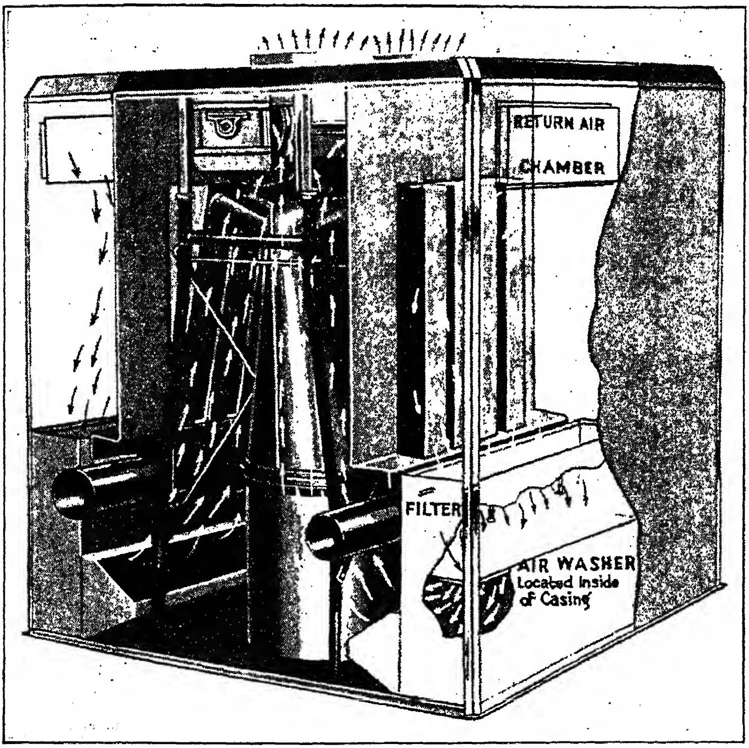


Fig. 58. Typical Gas-Fired Mechanical Hot-Air Furnace
Courtesy of Dail Steel Products Company, Lansing, Michigan

is forced by a blower or fan. This makes for more positive control of volumes and velocities and enables all rooms to be heated, despite their location. The mechanical system lends itself more readily to automatic control and allows a much more even temperature and relative humidity. An installation advantage is that the furnace can be placed anywhere in the basement and the ducts may be rectangular in section and smaller in area.

The essential difference between gravity and mechanical systems is the addition of a blower (centrifugal fan) which causes the air to circulate under pressure and thus reach all rooms in ample amounts. The blower is installed in many different ways, depending upon the

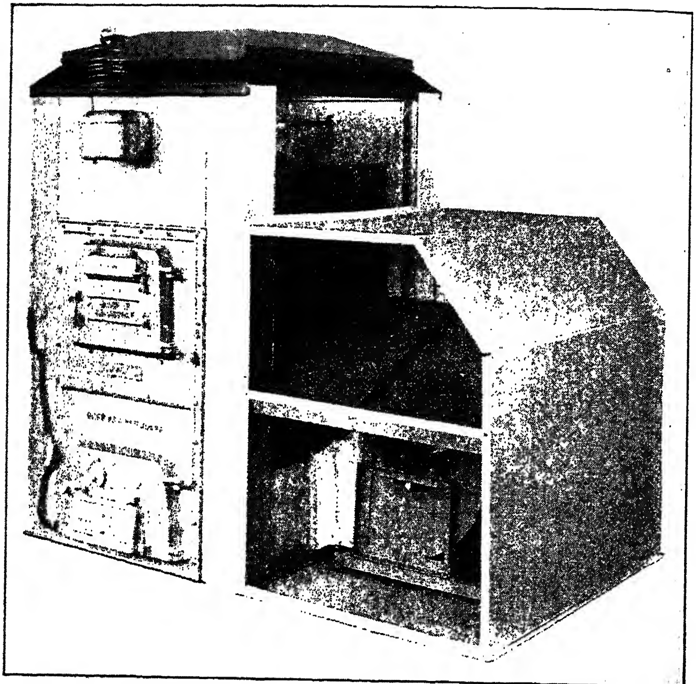


Fig. 59. Mechanical Furnace for Winter Air Conditioning
Courtesy of The Henry Furnace and Foundry Co., Cleveland

basic principle of construction. In all cases the first objective is to cause the air to circulate in a positive manner.

Fig. 58 shows a typical gas-fired, mechanical hot-air furnace where the blower is installed at the top of the furnace. This furnace functions as follows. The blower draws in the return air from the cold-air return pipes and draws it downward through the conditioning apparatus as shown by arrows. From the conditioner, the air passes through and around the furnace radiator where it *wipes* or *scrubs* the heating surfaces and is heated in the process. It is then blown out through the top of the furnace as shown by arrows. The ducts or leaders are attached at this point.

Fig. 59 shows another type of mechanical furnace. This is a hand-fired coal furnace although any of the automatic stokers or oil or gas burners of any standard type could be used. The furnace itself operates on the forced-air principle. The air is taken either from the return pipes or from the outside, and first caused to

Table 48

Model Numbers	Power Supply				Oil Rate	Max. Power Fan Motor (Input)	Duct Pressure	Tip Letter	Air Flow	Rated Heating Capacity
	Hp.	Volts	Phase	Cycles	G.p.h.	Watts	In. Water	—	C.f.m.	B.t.u. per hr.
21LB3A1	1/6	110	1	60	0.92	290	0.1	C	1,040	100,000
	1/4	110	1	60	0.92	370	0.2	C	1,040	100,000
21LB3B1	1/6	110	1	50	0.92	285	0.1	C	1,040	100,000
	1/4	110	1	50	0.92	370	0.2	C	1,040	100,000
21LB3C1	1/6	110	1	25	0.92	285	0.1	C	1,040	100,000
	1/4	110	1	25	0.92	370	0.2	C	1,040	100,000

Table 49

Model Numbers	Air Heating Capacity			Delivery Temperature at Unit			Humidifying Capacity
	Without Humidifier and Hot Water Coil	With Humidifier Only	With Humidifier and Hot Water Coil	Without Humidifier and Hot Water Coil	With Humidifier Only	With Humidifier and Hot Water Coil	
	B.t.u. per hr.	B.t.u. per hr.	B.t.u. per hr.	Deg. F.	Deg. F.	Deg. F.	Lb. per Hr.
21LB3A1 21LB3B1 21LB3C1	100,000	97,250	90,250	159.0	156.5	150.3	2.5

Table 50

Model Number	Output* B.t.u. per hr.	Input* B.t.u. per hr.	Humidifying Lb. per hr.	Fan Characteristics	
110 Volts 60 Cycles				Air Flow C.f.m.	Duct Pressure Inch Water
21LG1A1	35250	47000	2	500 400	0.1 0.2
21LG2A1	70500	94000	4	1000 900	0.1 0.2
21LG3A1	105750	141000	6	1300 1125	0.1 0.2
21LG4A1	141000	188000	8	1600 1350	0.1 0.2

*Rating Conditions: Entering air dry-bulb, 70° F. Entering air relative humidity, 35 per cent. Barometric pressure, 29.9 in. mercury. In accordance with American Gas Association Ratings.

pass through the filters, where dust and other suspended matter is filtered out. From the filters the blower forces air through the furnace, where by coming in contact with the radiator it becomes heated. Before leaving the bonnet or top of the furnace, the air is

humidified by an arrangement similar to the Zephyr Humidifier shown in the section on Humidification. Thus the air is cleaned, heated, humidified, and circulated.

Fig. 60 shows a somewhat similar arrangement to that shown in Fig. 59 with the addition of an automatic stoker. However, Fig. 60 more clearly illustrates how the mechanical principles can be ap-

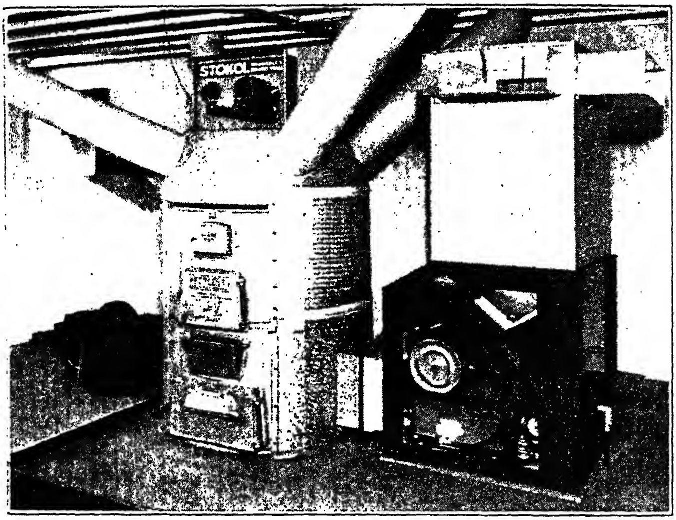


Fig. 60. Typical Mechanical Furnace System
Courtesy of Schwitzer-Cummins Co.

plied to the more familiar round gravity furnace. The return or cold air must be concentrated at one point. Rectangular ducts accomplish this. The cold air is then forced through filters and through the furnace as described for Fig. 59.

Fig. 61 shows an oil-fired type of mechanical furnace having the fan located within the unit. The cool air is drawn off the floors of the various rooms by means of the fan and return ducts and is filtered as it enters the unit. After being heated by passing through the heat-transfer part of the furnace, shown on the left-hand side of Fig. 61, the air is humidified, and sent out through the ducts.

The burner unit consists of a motor-compressor and a hermetically sealed burner head. The compressor is mounted above the fan assembly and the burner-head above the combustion chamber. The

burner head is sealed against leaks at all joints with leak-proof solder.

Tables 48 and 49 give typical data and ratings for warm-air conditioners of the type shown in Fig. 61.

Fig. 62 shows another typical mechanical furnace of the type enclosed within one unit casing. The openings at the top are inlet

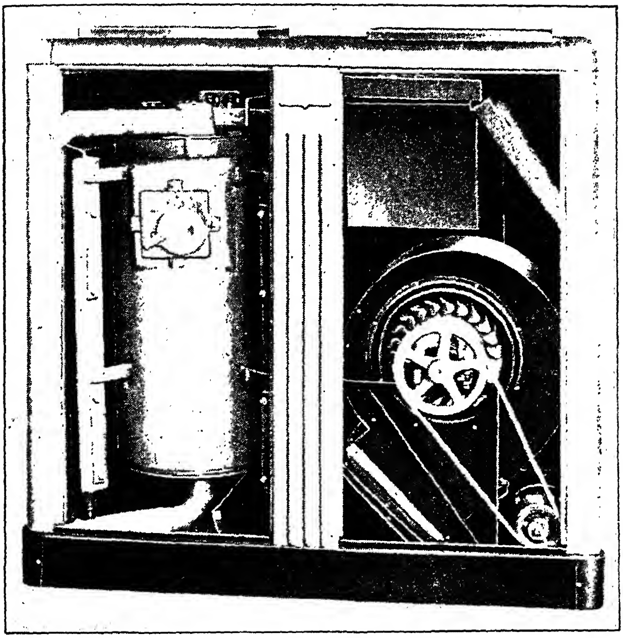


Fig. 61. Oil-Fired Warm-Air Conditioner
Courtesy of General Electric Company

and outlet openings. This apparatus consists of combustion heat-transfer units, suitable for gas burners, a radial flow fan, and air-conditioning parts. Table 50 shows ratings for a unit such as shown in Fig. 62.

Fig. 63 shows a type of warm-air conditioner in which oil is used as fuel. The burner principle is that of atomization.

The air, in the burning process, is supplied for atomization and primary combustion through the burner nozzle. For secondary combustion, the air is admitted through a refractory nozzle placed in the bottom of the combustion chamber. The secondary air directs

the flame upward so that it curls back on itself, giving a quiet, efficient, floating type of flame. As in Fig. 62 the circulation of heated air is entirely mechanical, being forced by a fan. These two types of mechanical warm-air furnaces shown in Figs. 62 and 63, can be used in new residences or small stores, etc., but not in structures where leaders are used, unless a new duct layout is made so

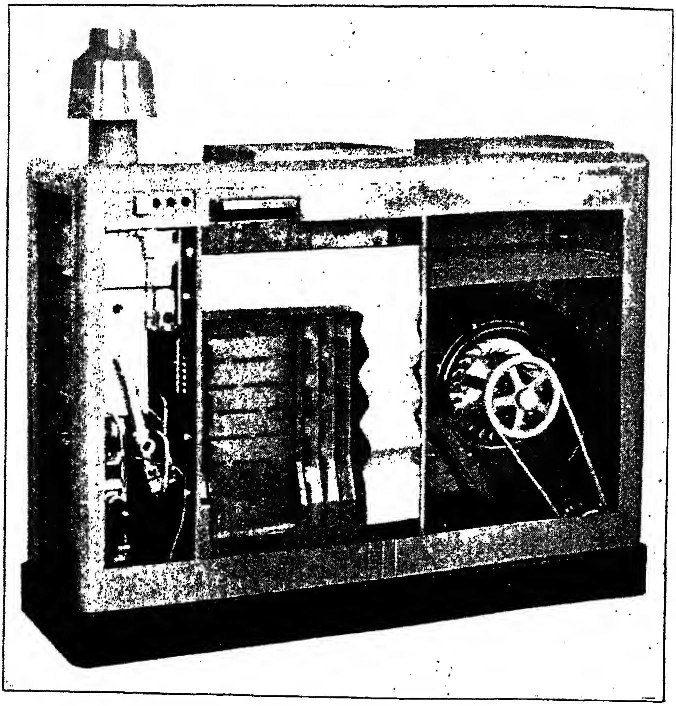


Fig. 62. Gas-Fired Warm-Air Conditioner
Courtesy of General Electric Company

that all ducts originate in one central duct which is attached to the furnace. The use of oil and gas-fired furnaces makes automatic control much more positive than where coal is used as fuel, except where automatic stokers are used.

Fig. 64 shows a typical residence and a mechanical hot-air furnace. Here the blower D is alongside of the furnace A and the two parts are connected by a passage X at the floor level. In this system the return air is drawn through cold air ducts C to the blower

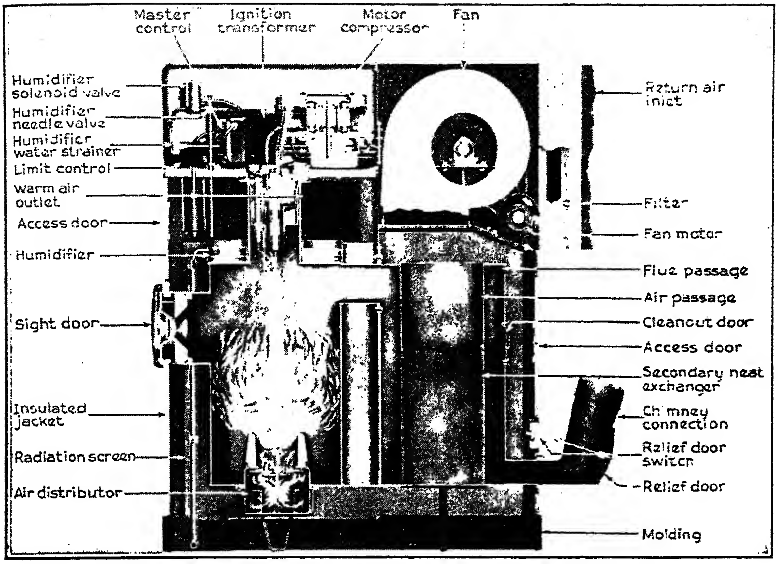


Fig. 63. Vertical Section of Typical Oil-Fired Warm-Air Conditioner
Courtesy of General Electric Company

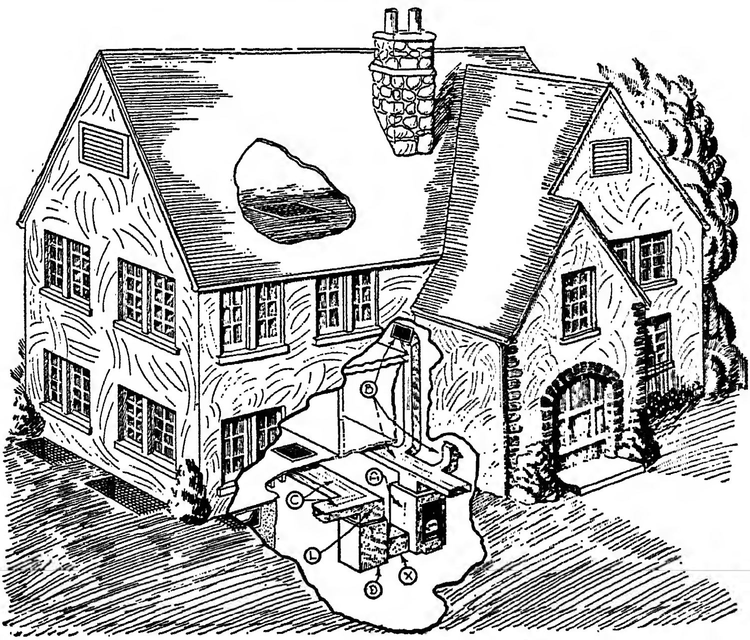


Fig. 64. Section of Typical Residence Showing Installation of a Mechanical Furnace System

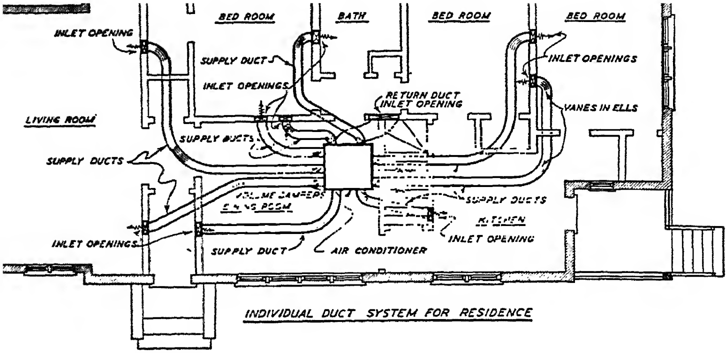


Fig. 65. Air Supply System for Residence Showing Individual Ducts

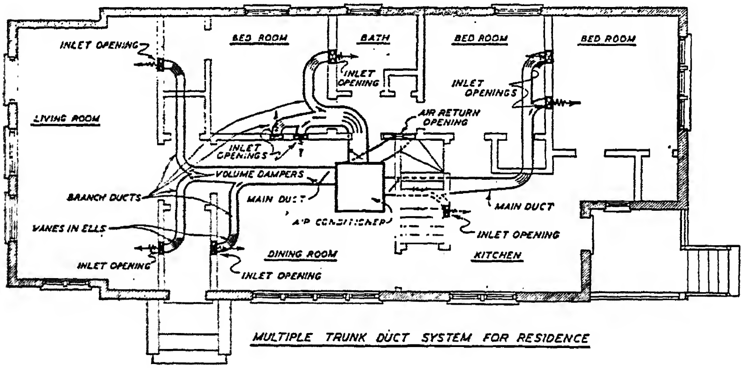


Fig. 66. Duct System for Residence Showing Multiple Trunk Ducts

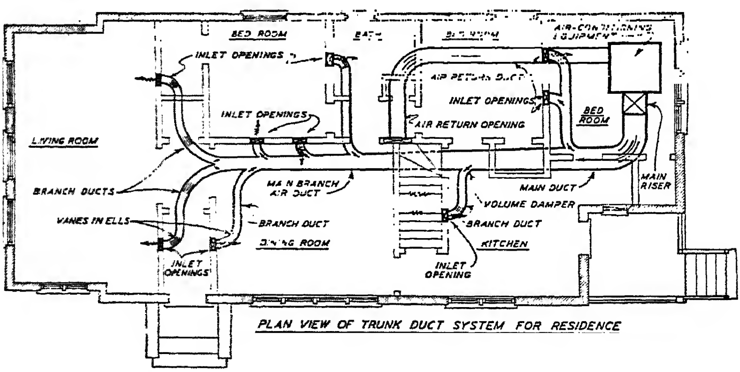


Fig. 67. Duct System for Residence Showing Trunk Ducts

D after having passed through the filter at L. From D the fan blows the air through X and into and through the furnace A where it is heated. From the furnace the air proceeds through ducts or leaders B to the various registers.

From the foregoing explanation it can be seen that no matter where the blower is located, the same results are obtained.

Duct Systems. There are three common ways in which ducts or leaders are installed, namely, individual ducts, multiple trunk ducts, and trunk ducts, as shown in Figs. 65, 66, and 67.

Note: In Figs. 65, 66, and 67, the furnace, blower, etc., are shown as a square labeled "Air Conditioner." This simplifies the drawings where the main interest lies in the ducts.

In the individual duct system, Fig. 65, a separate duct runs from the furnace to each room. The return air is taken care of by one centrally located return duct or cold-air pipe.

In the multiple trunk duct system, Fig. 66, there are three main ducts from which the individual ducts run to each room. The return air is taken care of by a single centrally located return duct or cold-air pipe.

In the trunk duct system, Fig. 67, the arrangement of ducts is similar to the multiple trunk duct system except that the main ducts are longer, which makes the individual ducts to each room shorter. Also, in trunk duct systems fewer main ducts are generally used and the furnace can be placed out of the way in one corner of the basement.

*THE TECHNICAL CODE FOR THE DESIGN AND INSTALLATION OF MECHANICAL WARM-AIR HEATING SYSTEMS

Intent of the Code: This Code is formulated to provide reliable methods of design and installation of mechanical warm-air heating systems for the protection of the buying public.

It covers fundamental data and rules which of necessity must be observed. In all cases, however, *experience* is necessary to obtain the full value of its application. *This Code does not attempt to provide the application of all variables found in practice.*

ARTICLE 1

SUGGESTED FIELD OR SCOPE OF LIMITATIONS FOR THIS CODE

Capacity in B.t.u. per hour = 250,000 at the registers.

For larger or unusual systems it is advisable to employ more exact methods to insure good results.

Cubic feet of air moved per minute (c.f.m.) = 5500

Height not exceeding three-story structures

Temperatures. Inside, as desired, usually 70° F. (min.): outside, See Table 55. Register air temperatures recommended for design: 110° F. (min.), 150° F. (max.)

ARTICLE 2

DEFINITIONS

Section 1. *Mechanical warm-air heating systems,* to which this code refers, shall consist of one or more warm-air heating units within individual housings or within one common housing, one or more motor-driven blowers, smoke or vent pipes, individual leader pipes or trunk line systems, or both, with necessary control dampers for supply and return air, automatic controls, registers, faces and grilles; and with provision for other appurtenances such as filters, air washers, ozonators, humidifiers, etc., as may be desired.

Section 2. *An individual supply system* shall consist of separate ducts of round or rectangular cross sections, without branches, and extending from the heating unit. Return ducts may be individual or trunk lines or a combination of both.

Section 3. *A trunk line system* consists of one or more main ducts with branches.

ARTICLE 3

ESTIMATING THE HEAT LOSS FROM BUILDINGS†

Section 1. This is done by finding the heat loss, H, (in B.t.u. per hour)

*Courtesy of National Warm-Air Heating and Air Conditioning Association.

†The symbols for heat loss shown in the Code are slightly different than explained in previous text material. The Code symbol for example, H_b is used for total heat loss whereas this text uses symbol H.

In like manner, formulas used to find loss through various structural parts are different from those shown in other parts of the text. However, this need not cause any trouble because, if desired, the heat loss can be determined exactly as explained on pages 78 and 79. Formulas (29) to (40) are herein reproduced as they appear in the Code.

from each room and then combining all the heat losses for the several rooms into one heat loss, H_b, for the building. The values of H are to be used in estimating the sizes and capacities of the *registers*, *grilles*, *stacks*, and *supply* and *return ducts* for the respective rooms; and the value H_b is to be used in estimating the sizes and capacities of the various parts of the combined heating system for the building, composed of *furnaces*, *blowers*, *washers*, *filters*, etc.

In estimating the heat losses for the various rooms, several items necessarily combine to form the value H; i.e., losses by transmission through (1) exposed glass, (2) exposed wall (including basement wall above grade), (3) partition wall to adjoining rooms of lower temperatures, (4) ceilings to attic spaces, (5) ceilings, part of roof, no attic space, (6) floors to unexcavated or cold spaces below, (7) basement walls below grade to surrounding ground, (8) basement floors to the supporting ground underneath; and (9) by infiltration. These are calculated as follows:

FORMULAS

Glass loss	B.t.u. per hour $= U_g A_g (t_i - t_o)$	(29)
Exposed wall loss (net area)	B.t.u. per hour $= U_w A_w (t_i - t_o)$	(30)
Loss through partition walls	B.t.u. per hour $= U_p A_p (t_i - t_p)$	(31)
Loss through ceilings	B.t.u. per hour $= U_c A_c (t_c - t_a)$	(32)
Loss through ceilings, part of roofs	B.t.u. per hour $= U_r A_r (t_c - t_o)$	(33)
Loss through floors above unexcavated spaces or to cold rooms below	B.t.u. per hour $= U_f A_f (t_f - t_s)$	(34)
Loss through basement walls below grade to surrounding ground	B.t.u. per hour $= U_x A_x (t_i - t_g)$	(35)
Loss through basement floors on ground	B.t.u. per hour $= U_y A_y (t_f - t_g)$	(36)
Infiltration loss	B.t.u. per hour $= 0.018CL(t_i - t_o)$	(37)
Outside air to replace vented air	B.t.u. per hour $= 0.018(\text{c.f.m.})60(t_i - t_o)$	(38)

H (for each room) = heat loss as given by the sum of the results obtained from Formulas (29) to (38) (39)

H_b = Total H values obtained for the building (40)

EXPLANATION OF TERMS IN FORMULAS (29) TO (38)

Section 2.

Formula (29) U_g = unit rate of heat transmission through the glass; A_g = area of exposed glass surface in square feet; t_i = inside design air temperature; t_o = outside design air temperature as per Table 55.

Formula (30) U_w = unit rate of heat transmission through the exposed wall; A_w = area of net exposed wall surface in square feet; t_i = inside design air temperature; t_o = outside design air temperature as per Table 55.

Formula (31) U_p = unit rate of heat transmission through the partition; A_p = area of the partition surface in square feet; t_i = inside design air temperature; t_p = air temperature in cold room or enclosed space. See Table 51.

Formula (32) U_c = unit rate of heat transmission through the ceiling construction; A_c = area of ceiling surface in square feet; t_c = temperature of the room air at ceiling (usually 1° per foot above the breathing level, in residences); t_a = air temperature in attic or space above ceiling. See Table 51.

Formula (33) U_r = unit rate of heat transmission through the roof to the outside air; A_r = area of roof surface in square feet; t_c = temperature of the indoor air near the roof; t_o = outside design air temperature as per Table 55.

Formula (34) U_f = unit rate of heat transmission through the floor to the unheated space; A_f = area of floor surface in square feet; t_f = temperature of the air at the floor level (usually 5° below room design air temperature); t_s = air temperature within the unheated space. Temperatures of unoccupied spaces below first floor, with tight floor and wall, usually taken as an average between inside and outside design air temperatures.

Formula (35) U_x = unit rate of heat transmission through the basement wall below grade to surrounding ground; A_x = area of basement wall surface below the grade line in square feet; t_i = design air temperature of the basement room; t_g = assumed average temperature of the surrounding ground (may be taken as 25° F. in zero weather).

Formula (36) U_y = unit rate of heat transmission through the floor to the ground; A_y = area of floor surface in square feet; t_f = temperature of the room air at the floor; t_g = assumed temperature of ground below the floor (may be taken 20° below room design air temperature).

Formula (37) C = air leakage, Table 54, in cubic feet per lineal foot of crack per hour at 15 m.p.h. wind velocity. L = length of exposed crack in feet. $0.018 = 0.24 \times 0.075$. In a room with one exposed wall, use the total length of crack. In a room with two exposed walls use the side having the greatest length of crack. In a room with three or more exposed walls take the wall having the greatest length of crack, but in no case take less than one-half the total crack. t_i = inside design air temperature; t_o = outside design air temperature as per Table 55.

Formula (38) See Art. 6, Section 2, Item (a) 7th par.

Section 3. **Table 51**

U_c or U_p	t_a or t_p When Out-of-Door Temperature is		
	0°	−10°	−20°
.6 B.t.u. per hour	35	30	26
.5 B.t.u. per hour	32	27	23
.4 B.t.u. per hour	28	24	20
.3 B.t.u. per hour	23	20	17
.2 B.t.u. per hour	18	16	14
.1 B.t.u. per hour	10	9	8

The temperature t_a or t_p on the unheated side of a ceiling or a partition next to an unfinished attic space varies with the construction. Under ordinary conditions it can be assumed: when the rates of heat transmission (U_p) or (U_c) are as in Table 51, the corresponding cool side air temperatures (t_a) or (t_p) will prevail.

Section 4.

Table 52. Recommended Conductivities and Conductances for Computing Heat Transmission Coefficients

Conductivities (k) are expressed in B.t.u. per hour per square foot of surface per degree difference (Fahrenheit) per inch thickness; $\frac{1}{k}$ is *resistivity* on the same basis. *Conductances* (C) are indicated by asterisks (*) and are for thickness or condition stated, *not* per inch; $\frac{1}{C}$ is *resistance* on the same basis.

Note: Table 52 is not reproduced here because the same values are shown in Table 2, Chapter V, Vol. II

Table 53. Heat Transmission Coefficients for Average Building Constructions

Section 5. The heat *transmission coefficients* given below represent the unit rate of heat transmission or B.t.u. heat loss per square foot of surface per hour per degree difference between the temperatures on the heated and the unheated sides of the surface.

Outside walls and roofs are based on 15 mile per hour wind.

Note 1. Table 53 is not reproduced here because it is exactly the same as Tables 3 to 13 in Chapter V, Vol. II.

Note 2. Tables 3 to 13, Chapter V, Vol. II, show U values of various walls, floors, roofs, etc. But according to Formulas (29) to (38), the values U_w, U_f, U_c, etc. are required. Therefore assume that these Code symbols mean the same as U.

Example: U of glass in Table 13, in Chapter V, Vol. II, is 1.13. Thus in the Code symbols U_g equals 1.13.

Note 3. For infiltration the U value is taken as .018 per cubic foot.

Table 54. Infiltration

Section 6. Expressed in Cubic Feet per Foot of Crack per Hour.

Note 1. This table is not reproduced because Table 19, Chapter V, Vol. II, is exactly the same.

Note 2. The following values will be found useful. They are expressed in the same terms as in Table 19, Chapter V, Vol. II.

Doors, well fitted	100
Doors, poorly fitted	150
Doors, weatherstripped	50

Section 7. The following table lists the recommended outdoor winter temperatures to use as the basis for preparing heating estimates. These figures should be used unless the customer specifies the temperature upon which the estimate shall be made.

The figures given in this table are not the extreme minimum temperatures recorded in each locality and it is to be expected that lower temperatures will be encountered. However, very low temperatures usually exist only for short periods. It is thus unnecessary and uneconomical to design the heating system for the extreme minimum temperature.

Table 55. Outside Air Temperatures for Heating Estimates*

State-City	Design Temp. F.
ALA.	
Birmingham	5
Mobile	15
ARIZ.	
Phoenix	25
Tucson	20
ARK.	
Fort Smith	10
Hot Springs	5
CALIF.	
Los Angeles	30
Pasadena	25
San Diego	35
San Francisco	35
COLO.	
Denver	−20
Pueblo	−25
CONN.	
Hartford	−5
New Haven	−5
DEL.	
Wilmington	5
D. of C.	
Washington	0
FLA.	
Jacksonville	25
Miami	35
Tampa	35
GA.	
Atlanta	10
Columbus	10
IDAHO	
Boise	0
ILL.	
Chicago	−10
Danville	−5
Peoria	−10
Springfield	−10
IND.	
Evansville	0
Indianapolis	−5
IOWA	
Cedar Rapids	−5
Davenport	−10
Des Moines	−15
KAN.	
Leavenworth	−10
Topeka	−10
KY.	
Frankfort	0
Louisville	0
LA.	
New Orleans	25
Shreveport	15
ME.	
Bar Harbor	−15
Portland	−10
MD.	
Annapolis	5
Baltimore	5
MASS.	
Boston	0
Concord	−15
Springfield	−10
MICH.	
Ann Arbor	−5
Detroit	0
Grand Rapids	−10
Sault Ste. Marie	−20
MINN.	
Duluth	−25
Minneapolis	−20
MISS.	
Jackson	15
Vicksburg	15
MO.	
Kansas City	−5
St. Louis	−5
MONT.	
Butte	−20
Helena	−25
NEB.	
Lincoln	−10
Omaha	−10
NEV.	
Las Vegas	20
Reno	−5
N. H.	
Concord	−20
N. J.	
Atlantic City	10
Trenton	5
N. M.	
Albuquerque	5
Santa Fe	0
N. Y.	
Albany	−5
Buffalo	−5
New York City	0
Syracuse	−15
N. C.	
Asheville	5
Raleigh	15
N. D.	
Bismarck	−30
OHIO	
Cleveland	−5
Dayton	−10
Toledo	−5
OKLA.	
Oklahoma City	−5
ORE.	
Baker	0
Portland	15
PA.	
Erie	0
Philadelphia	5
Pittsburgh	−5
R. I.	
Providence	0
S. C.	
Charleston	20
S. D.	
Aberdeen	−30
Sioux Falls	−20
TENN.	
Knoxville	10
Memphis	5
TEXAS	
Dallas	10
Houston	25
San Antonio	20
UTAH	
Salt Lake City	−5
VT.	
Burlington	−10
VA.	
Norfolk	15
Richmond	10
WASH.	
Seattle	10
Spokane	−10
W. VA.	
Charleston	0
Wheeling	−5
WIS.	
Madison	−10
Milwaukee	−10
WYO.	
Cheyenne	−20

*Courtesy of Industrial Press—Publishers of Heating and Ventilating.

METHOD OF CALCULATING (U)

Section 8. For types of construction not given in Table 53, the value of U may be obtained from Formulas (2) through (7), Chapter V, Vol. II.

Note: The above method is given in the explanation of the formulas.

ARTICLE 4

SELECTION OF THE FURNACE

Section 1. The furnace shall be selected to satisfy H_b of building.

The furnace used for burning solid fuels shall be rated as follows:

$$\text{B.t.u. at register} = .85(G \times E \times F \times C)[1 + 0.02(R - 20)] \tag{41}$$

Where G = grate area in square feet; E = efficiency of the furnace at the bonnet, (.65); F = calorific value of the fuel (unless otherwise specified, manufacturers generally use 12,000 B.t.u. for catalogue ratings); C = combustion rate, i.e., pounds of fuel burned per hour per square foot of grate (7.5), with a minimum draft at the smoke-pipe connection to the furnace of not less than .12 inches of water; and R = ratio of heating surface area to grate area.

For furnaces designed to burn fuel oil or gas, use manufacturers' ratings. In no case shall the input exceed the furnace manufacturers' specifications. It is not advisable to use more than 75 per cent bonnet efficiency in sizing equipment. See Table 50.

When furnaces are stoker fired use a combustion rate of not to exceed

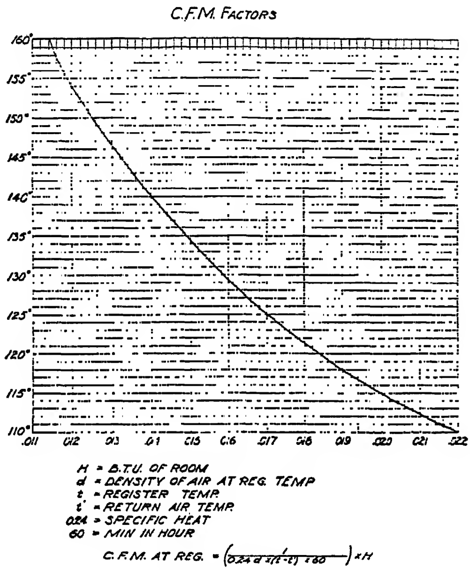


Fig. 68. C.F.M. Factors

10 pounds per square foot of equivalent grate area per hour for normal operation, and 65 per cent efficiency at the bonnet for 12,000 B.t.u. coal.

For highest efficiencies, when using blowers, baffling is recommended.

ARTICLE 5

REGISTER AIR TEMPERATURE TO BE USED IN DESIGN

Section 1. To find the total c.f.m. for the rooms to be heated multiply the total cubic contents of rooms by the number of air changes selected and divide by 60. The total building c.f.m. *requirements* shall be equal to, or greater than, five complete air changes through the system per hour, dependent upon the resultant register air temperature (See Section 2). To determine the average register temperature, divide the total building c.f.m. by H_b for c.f.m. factor and refer to Fig. 68. Systems installed in old houses utilizing old stacks are exceptions.

Section 2. Fig. 68 is charted by use of the following formula;

$$\text{C.f.m. at Register} = \left(\frac{1}{0.24d \times (t-t') \times 60}\right) \times H \tag{42}$$

H = B.t.u. of room; d = density or weight of entering air at register in pounds

per cubic foot; t = temperature of entering air (register temperature); t' = temperature of air leaving room (return air temperature). 0.24 = specific heat of air, or B.t.u. required to raise one pound of dry air one degree F.; 60 = min. per hr.

For the usual range of entering air temperatures the terms written in the bracket in Formula (42) may be reduced to a single factor and Formula (43) may be written:

$$\text{C.f.m.} = \text{Factor} \times H \tag{43}$$

These factors, corresponding to the various register air temperatures and based on 65° F. return air temperature, as given in Fig. 68, offer a simple and reliable method for obtaining the c.f.m. for each room.

Example—Suppose the register air temperature is 135° F., then from Fig. 68 the factor is 0.0149. If the heat loss (H) is 15,000 B.t.u. per hour, then the volume of air flowing to the room in c.f.m. = 0.0149×15,000 = 224.

It is common practice to proportion the return ducts upon the same volume basis as the supply ducts.

Section 3. C.f.m. requirement per room.

Due to the fact that the drop in temperature between the bonnet and the register is dependent upon the actual length of the run, it is necessary to compute the bonnet temperature (See Art. 6, Sec. 2, Item d) and establish the register temperature for each run (See Art. 6, Sec. 2, Item e). Multiply the B.t.u. load of each run by the factor corresponding to the register temperature, as shown in Fig. 68.

ARTICLE 6

DUCT DESIGN PROCEDURE

Section 1. *Outline.*

(a) Make a line diagram of tentative layout. See Fig. 69.

(b) Measure the actual length of each run from register to furnace. See Art. 6, Sec. 2, Item (b).

(c) Compute the equivalent length of each turn in each run from register to furnace and add the same to the actual length of the respective runs. These results are the total equivalent lengths. See Art. 6, Sec. 2, Item (c).

(d) Compute the bonnet air temperature. See Art. 6, Sec. 2, Item (d).

(e) Compute the register temperature of each run. See Art. 6, Sec. 2, Item (e).

(f) Compute the c.f.m. for each register based on warm-air temperature at each register. See Art. 6, Sec. 2, Item (f).

(g) Select the run having the greatest friction loss and base the design on this resistance. See Art. 6, Sec. 2, Item (g).

(h) Determine round pipe size for each run. See Art. 6, Sec. 2, Item (h), also Table 56.

(i) Correct pipe diameter of each run for various equivalent lengths. See Art. 6, Sec. 2, Item (i).

(j) Compute rectangular equivalents for stack and branch sizes. See Art. 6, Sec. 2, Item (j), also Tables 57 and 58.

(k) Determine trunk sizes in round pipe diameters. See Art. 6, Sec. 2, Item (k).

(l) Correct round pipe diameters for each trunk. See Art. 6, Sec. 2, Item (l).

(m) For rectangular equivalents of trunk sizes, see Art. 6, Sec. 2, Item (m).

(n) Follow same procedure for the design of the return side of the system as is used for the warm air side. See Art. 6, Sec. 2, Item (n).

(o) Compute total resistance of the system. See Art. 6, Sec. 2, Item (o).

(p) SELECT BLOWER to supply the total c.f.m. at a static pressure equal to or greater than the total resistance of the system. See Art. 6, Sec. 2, Item (p). SELECT MOTOR TO SUIT BLOWER.

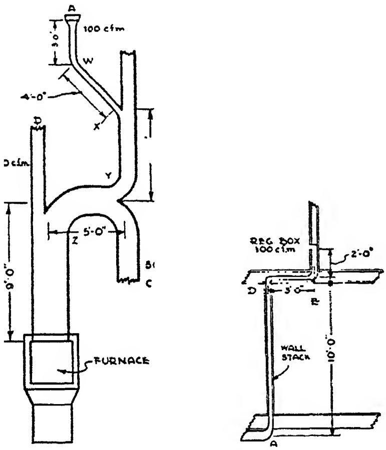


Fig. 69. Typical Duct Diagrams

(q) Select filters, washers and other appurtenances. See Art. 6, Sec. 2, Item (q).

(r) Select registers and grilles. See Art. 6, Sec. 2, Item (r).

(s) Select controls. See Art. 7.

Section 2. Explanatory Notes, relating to Section 1.

Item (a) A general knowledge of building construction is necessary to aid in an intelligent and practical heating layout.

If a room requires more than 150 c.f.m., it is best to use two or more supply openings to bring about better room circulation.

It is advisable to oversize recreation room supply to counteract radiant loss to bare basement walls and pickup load due to intermittent heating. This may be done by using a design room temperature from 5 to 10 degrees above that for living rooms.

It is advisable, when possible, to install a separate run direct from heater unit to garage, should same be heated. It is also advisable to exercise extra care in sizing supply and return lines to rooms over garages.

Return air shall be brought back from as many rooms as possible excepting from bathrooms and garages.

Care should be taken to draw from each room or series of intercommunicating rooms air equivalent to the amount designed to be delivered to the room or series of rooms; also that the *total* air to be returned shall in no case be less than the *total* equivalent supply of warm air to all rooms, unless compensated for by the admission of outside air.

It is suggested that equivalent air be introduced into the system from the outside to compensate for that vented from garages. A little outside air (approximately 5% by volume) is a good practice on all jobs.

Item (b) Actual Length.

The actual length of a rectangular or round warm-air or return-air run is the lineal feet of pipe or duct between the register and the furnace.

Example 1. See Diagrams A and B, Fig. 69. Add lineal feet of stack from the register box along the line E, D, A, W, X, Y, Z, to furnace, equalling 42 feet, which is the actual length of that run.

Item (c) Equivalent Length.

The equivalent length of a round or rectangular run is the total actual length, plus the equivalent lengths in straight pipe or duct, of all turns.

Assume all 90° turns equal to 10 feet of straight pipe or duct, and 45° turns and register boxes equal to 5 feet of straight pipe or duct. See Table 59 if more accuracy is desired.

Example 2. See Fig. 69. Add equivalent length of register box to turns at E, D, A, W, X, Y, Z, and at furnace, equalling 75′. Add to this the actual length of the run, 75′+42′=117′= total equivalent length of the run.

Item (d) To determine bonnet temperatures.

In order to compensate for temperature drops, between furnace and registers, it is necessary to compute the temperature of air at each register. From data collected in the field, it is safe to assume a temperature drop of ¼ of 1 degree per foot under ordinary conditions. In Art. 5, Sec. 1, an average register temperature was arrived at by dividing the total building c.f.m. by H_b and referring to Fig. 68. This average register temperature represents the design temperature of a register located at a point one-half the actual length of the longest run.

Example 3. If the actual length of the longest run is 60 feet, the average register temperature is the design temperature of a register 30 feet from the furnace.

Example 4. If the longest run is 150 feet, the average register temperature is the design temperature at 75 feet.

Inasmuch as there is an assumed temperature drop of ¼ of 1 degree per foot, the bonnet temperature is the average temperature plus ¼ times ½ the lineal feet of the longest run.

Example 5. Referring to Example 3 where the longest run was 60 lineal feet, the bonnet temperature would be the design temperature $+\left(\frac{1}{4}\times\frac{60}{2}\right)$. Assuming an average design temperature of 135° F., the bonnet temperature would be $135+\left(\frac{1}{4}\times\frac{60}{2}\right)=135+7.5=142.5°$ F.

Item (e) Having determined bonnet temperature, the correct air temperature at any register may be computed by deducting from the bonnet temperature ¼ of 1 degree for each lineal foot between the bonnet and that register.

Example 6. Where the bonnet temperature is 142.5 F. and the register is 40 feet from the bonnet, the register temperature is 142.5°−(¼×40)=142.5°−10=132.5° F.

By the above method, compute the correct temperature at each register.

Item (f) In order to compute the correct c.f.m. on each run, multiply the B.t.u. load of each run, by the factor obtained from Fig. 68 corresponding to the correct register temperature.

Example 7. Example 6 shows the air temperature of a certain register to be 132.5° F. From Fig. 68 we find the c.f.m. factor corresponding to 132.5° F. to be .0154. If H, the B.t.u.'s to be delivered by that register, were found to be 6500, 6500×.0154 would equal 100.1 c.f.m. In like manner find c.f.m. of any warm-air run.

Item (g) Selection of Design Friction Loss.

The *total* resistance to flow of air through ducts is due to two types of losses—friction losses and dynamic losses. This is equivalent to the friction losses in all straight runs, plus the dynamic losses in all elbows and turns expressed in terms of friction losses in equivalent length of straight duct.

The *commonly accepted friction chart* is shown in Fig. 70. In the use of this chart a fair recommendation would be to design the average systems, where the capacities of the trunk having the longest equivalent run requires as much as 1,600 c.f.m., on a resistance of .06 inch; from 1,600 to 2,400 c.f.m. on .08; from 2,400 to 3,600 c.f.m. on .10; and from 3,600 to 5,500 c.f.m. on .12 inch of water per 100 feet of equivalent length.

Where limitations of pipe size of any stack prevent the use of the above recommendations, the design static pressure may be arrived at in the following manner:

By observation select the run having the greatest resistance and find its total *equivalent* length. To determine this resistance, from Fig. 69, find the resistance in inches of water per 100 feet necessary to handle the required c.f.m. through this duct of available size. Multiply this resistance per 100 feet by the total equivalent length and divide by 100. The result is the design static pressure of this run, as well as the design static pressure of the entire system provided it is within practical limits.

Items (h) and (i)

For individual round pipes refer to Table 56 for diameter required to handle c.f.m. at design static. Correct for pressure differences due to unequal equivalent lengths by multiplying diameter of pipe by the correction factor corresponding to the total equivalent length from bonnet to register, as shown in Table 60.

Example 8. The run in Example 7 was found to require a capacity of 100 c.f.m. If the design static were chosen at .06, the pipe diameter from Table 56 is found to be 6.8″. If the register were found to have an equivalent length of 80 ft. from the furnace, the correct diameter can be found by multiplying 6.8″ by the correction factor corresponding to 80 ft. from Table 60. 6.8×.965 = 6.56, or say, 6.6″ in diameter. The above method will size and correct all warm-air runs giving proper pipe diameters of runs up to two hundred feet of equivalent length.

Item (j) Sizing wall stacks and branches. See Tables 57 and 58.

Example 9. Example 8 gave a pipe diameter of 6.6″ to deliver 100 c.f.m. 80 feet of equivalent length from the furnace and at a static of .06. From Table 57 (Conversion Table of Round Pipe Diameters to Wall Stack Sizes), we find that a 10×4, an 11×3½ or a 13x3 stack will be satisfactory. The nearest commercial stack size in this case is a 12x3½. In like manner and by the use of Table 58, a 6×7, 5×8, 5×9 or 4×10 rectangular duct will satisfy the above conditions.

Item (k) Sizing trunk lines, in round pipe diameters.

Starting at the extreme end of the trunk line and working toward the heater unit, add the c.f.m. of the first two or more branches joining at that point. Find the diameter of the trunk for that point from Table 56 using desired design static. To the c.f.m. at this point add the c.f.m. of the next branch joining the

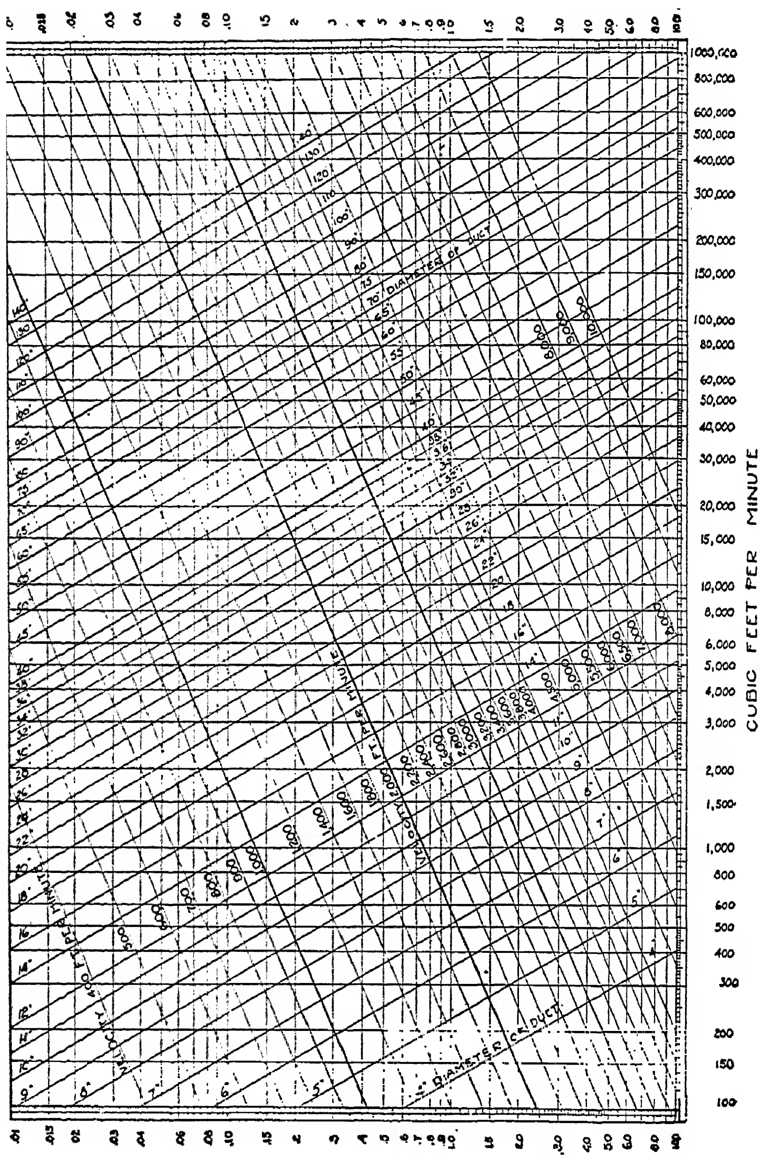


Fig. 70. Friction Chart

trunk and find the diameter of the trunk at this new location. In like manner add the c.f.m. of each succeeding branch to the c.f.m. of the duct at point of juncture.

Example 10. Analysis of one complete run. Referring to Diagrams A and B, Fig. 69, and Example 2, assume the run A to be the end run of the trunk line. This run must handle 100 c.f.m. and its equivalent length is 117 (use 120) feet. Assuming a design static of .06 the round pipe diameter is 6.8, see Table 56. From Table 60 (correction table) 6.8 ×1.035 = 7.038 (say 7.1) inches, the corrected round pipe diameter of the branch.

Adding the 300 c.f.m. of the first branch *B* to join the 100 c.f.m. of the branch *A* we find a total of 400 c.f.m. From Table 56, 400 c.f.m. at .06 static gives a pipe 11.4″ in diameter. 11.4 ×1.035 = 11.799 (say 11.8) inches, the corrected pipe diameter. Adding the 500 c.f.m. of *C* to the 400 c.f.m. of *A* and *B* we have 900 c.f.m. 900 c.f.m. at .06 static gives a pipe 15.5 inches in diameter. 15.5 ×1.035 = 16.04 (say 16.0″), the corrected pipe diameter. In like manner 200 c.f.m. at *D* plus 900 c.f.m. of $A+B+C$ = 1,100 c.f.m. 1,100 c.f.m. at .06 = 16.75″. 16.75 ×1.035 = 17.35 (say 17.4″), the diameter of trunk line at furnace.

Item (l) In order to correct the trunk line to bring about the design static regardless of its length, it is necessary to multiply the diameter of the trunk between branches by a common correction factor. This factor is the same factor used in correcting the diameter of the branch with the greatest equivalent length, regardless of the point of intersection with this trunk.

Should two trunk lines merge, treat each as individual trunks to point of intersection. Total c.f.m. of each trunk at this point and proceed as one trunk.

If there are two or more trunk lines to the furnace, design and correct each trunk as outlined above.

Item (m) Refer to Table 58 for rectangular equivalents.

Item (n) Size return-air trunks and branches, or individual ducts or pipes in the same manner as on the warm-air side of the system.

Item (o) Total resistance of entire system.

To find the total resistance of the entire system, it is necessary to add together the resistances offered by every portion of the entire system.

As the warm-air and return-air sides of the system are designed on a definite static pressure, and all runs corrected to offer this same static pressure, it is apparent that the design static of the circulating system is the sum of the two.

The *total* resistance against which the fan must operate is, therefore, the resistance of the circulating system, plus the resistance of that register on warm air and return having the greatest resistance, plus the resistances of the heater, filter (usually .05), washers, cooling coils, eliminators and other appurtenances.

Example 11. Assuming the design static of each system (warm air and return) to be .08, add the following:

.08 (warm-air side) + .08 (return-air side) + .03 (assumed static of register offering greatest resistance on warm-air side) + .02 (assumed resistance of register offering greatest resistance on return-air side) + .05 (resistance of furnace) + .10 (resistance of filters, partially dirty) = .36 static resistance against which the blower must operate.

Item (p) Selection of Blowers.

The blower capacity at 65° F. return air should be the equivalent of, or greater than, the sum of the c.f.m. required to supply all the outlets at the design warm-air temperatures, and at the total pressure of the entire system. It is assumed that all blower ratings are in accordance with the Standard Test Code of the A. S. H. & V. E. Guide.

Item (q) Filters, Washers and Cooling Coils.

It is recommended that all furnaces be provided with some method of cleaning the air. In the case of filters it is recommended that the maximum velocity through the filters should not exceed 200 feet per minute. Use manufacturer's ratings for resistances and efficiencies. Generous filter areas are advisable to counteract reduced capacities and increased resistances when dirty. When washers are used, use manufacturer's ratings as to capacities and resistance. When cooling coils are used, use manufacturer's recommendations as to velocities through coils and their ratings for resistance.

Item (r) Registers and Grilles.

Use registers and grilles of proper size and area. Same to be full width of stack. Use manufacturer's ratings for volume, velocities and areas as required.

When warm-air registers are placed in the baseboard, or just above the baseboard, they shall be sized on a basis of not more than 300 feet average velocity. Downward deflecting registers permit velocities up to 500 feet per minute.

When registers are placed above the breathing line (register top 18″ below ceiling), they shall be sized on a basis of 500 feet minimum velocity, excepting in bath rooms and toilets. Such registers must be horizontal or slightly downward direction of flow. Where the distance from the register to the opposite side of the room is over 15 feet, higher velocity should be used.

All baseboard or wall warm-air registers, either high or low, shall be properly sealed to the stack head or register box in such a manner as to prevent any leakage of air between the head and the register.

Back-pressure louver type registers must be used in garages.

The resistance through registers varies with design and velocity. Consult manufacturer's catalogue for data.

All return-air registers *should* be at the floor line.

ARTICLE 7

CONTROLS

Section 1. *Blower Controls.* An adjustable automatic furnace switch shall be placed in the bonnet of the furnace to start and stop the blower at predetermined temperatures. Under average conditions it should be set to start the blower at from 130 to 175° F., and to stop the blower at from 100 to 130° F. This setting should be determined by the register temperature at which the plant is designed to operate.

Section 2. *Summer Switch.* Unless automatically controlled, a manual switch for summer control may be placed in the hallway, stairway or other accessible location, but must not be placed where it is liable to be turned by mistake. This switch shall be so wired in conjunction with the automatic blower switch in the furnace bonnet that the blower cannot be stopped when the furnace is hot.

Section 3. *Limit Control.* A limit control shall be wired in conjunction with any automatic temperature control, to prevent overheating of the heater unit. This control shall be set at a temperature not higher than 250° F. The customary setting is not more than 30 degrees higher than blower-control setting.

Section 4. *Coal, Hand-Fired.* The control for a hand-fired coal heating

unit shall consist of a room thermostat operating an electric motor for opening and closing the furnace dampers. The same shall incorporate limit controls. See Section 3.

Section 5. *Coal, Stoker-Fired.* The controls for a stoker-fired heating unit shall consist of a room thermostat, limit control, a relay for operating stoker motor, and some form of hold-fire control, such as high-low stack switch or time interval contactor.

Section 6. *Oil-Fired.* Where the heating unit is oil-fired, use standard controls furnished with oil burner. The same shall include room thermostat, limit controls, pyrostat or other approved ignition control.

Section 7. *Gas-Fired.* When the heating unit is gas-fired, the controls shall consist of a room thermostat, limit control, and approved safety devices to close and vent the gas line.

Section 8. *Zone Controls.* The controlling of larger installations by zone controls is advisable. Such controls consist of thermostats operating dampers in warm-air ducts leading to certain portions of the residence. Unless familiar with the operation of zone controls, consult representative of manufacturer of control desired.

Section 9. It is imperative that the owner furnish a separate circuit of proper voltage and capacity to be run from the house service to a separate panel. This panel is for the *heating equipment only.*

ARTICLE 8

CONSTRUCTION DETAILS

Section 1. *Stacks.* Stacks shall be constructed of I.C. tin or 28-gauge galvanized iron. It is advisable to wrap all warm-air stacks with one layer of 10-pound asbestos paper. All joints shall be stripped with asbestos paper. Where stack heads, boots, or other fittings, either for warm air or return air go through the first floor, all openings around such fittings shall be filled with asbestos fiber or other noncombustible insulating materials to make this opening gas or dust tight. (Requirement of Fire Underwriters.)

Section 2. *Round Pipe Trunk Line.* Round pipe trunk line shall be constructed of galvanized iron: up to 14-inch, 26-gauge; to 18-inch, 24-gauge; larger than 18-inch, 22-gauge. If slip joints are used, joints shall be stripped with asbestos paper.

Section 3. *Rectangular Duct Trunk Line.* Rectangular ducts shall be constructed of galvanized iron: up to 12 inches wide, use 28-gauge; to 18 inches wide 26-gauge; to 30 inches wide, 24-gauge; and wider, 22-gauge. All ducts 24 inches or wider shall be crossbraced on top and bottom and shall have standing seams or angle-iron braces. All joints shall be S and drive strips, or locked. *No warm-air duct, round or rectangular, shall come in contact with masonry walls.* Insulate around warm-air ducts through masonry walls with not less than ½ inch of insulation.

Section 4. *Individual Round Pipes.* Individual round pipes up to 12 inches in diameter may be constructed of 26-gauge galvanized iron or I.X. tin.

Section 5. *Return-Air Liners.* Bottoms of all return-air joist spaces shall be lined with smooth air-tight material. Capacities shall be figured at 80 per cent of round pipe equivalent.

Table 56. C.F.M. Capacities of Individual Round Pipes

.06 c.f.m.	.08 c.f.m.	.10 c.f.m.	.12 c.f.m.	.15 c.f.m.	Rd. Pipe Size	.06 c.f.m.	.08 c.f.m.	.10 c.f.m.	.12 c.f.m.	.15 c.f.m.	Rd. Pipe Size
45	50	55	65	70	5.0*	460	535	600	650	750	12.0*
47	55	60	67	75	5.1	470	550	615	670	770	12.1
50	57	65	70	80	5.2	480	560	630	685	785	12.2
53	60	67	75	85	5.3	495	575	640	705	805	12.3
55	62	70	77	90	5.4	505	585	655	720	820	12.4
57	65	75	80	95	5.5	515	600	670	740	840	12.5
60	70	80	85	97	5.6	525	610	685	755	860	12.6
62	75	82	90	100	5.7	535	625	700	770	880	12.7
65	77	85	95	105	5.8	550	635	720	790	900	12.8
70	80	90	100	110	5.9	560	650	735	805	920	12.9
73	82	95	102	120	6.0*	570	660	750	820	940	13.0
75	85	100	105	125	6.1	580	675	765	835	960	13.1
78	90	102	110	130	6.2	595	690	780	850	975	13.2
80	95	105	115	135	6.3	605	710	800	870	995	13.3
85	100	110	120	140	6.4	620	725	815	885	1,010	13.4
90	105	115	125	145	6.5	630	740	830	900	1,030	13.5
92	110	120	130	150	6.6	645	755	850	920	1,050	13.6
95	112	125	135	160	6.7	660	770	865	935	1,075	13.7
100	115	130	145	165	6.8	670	780	885	955	1,095	13.8
105	120	135	150	170	6.9	685	795	900	970	1,120	13.9
110	125	140	155	175	7.0*	700	810	920	990	1,140	14.0*
112	130	150	160	185	7.1	714	825	935	1,010	1,160	14.1
115	135	155	170	190	7.2	730	840	950	1,030	1,180	14.2
120	140	160	175	200	7.3	740	860	970	1,050	1,200	14.3
125	145	165	180	205	7.4	760	875	985	1,070	1,220	14.4
130	150	170	185	215	7.5	770	890	1,000	1,090	1,240	14.5
135	155	175	190	220	7.6	785	905	1,020	1,110	1,265	14.6
140	160	180	195	230	7.7	800	920	1,040	1,135	1,290	14.7
145	165	190	205	235	7.8	810	940	1,060	1,155	1,310	14.8
150	170	195	210	245	7.9	825	955	1,080	1,180	1,335	14.9
155	180	205	220	255	8.0*	840	970	1,100	1,200	1,360	15.0
160	185	210	230	265	8.1	880	1,010	1,150	1,250	1,430	15.25
165	190	215	235	270	8.2	920	1,055	1,200	1,300	1,500	15.5
170	195	220	240	280	8.3	960	1,090	1,250	1,360	1,570	15.75
175	200	230	250	290	8.4	1,000	1,140	1,310	1,420	1,640	16.0*
180	210	240	260	300	8.5	1,040	1,190	1,370	1,480	1,710	16.25
190	220	250	270	310	8.6	1,080	1,245	1,430	1,540	1,780	16.5
195	225	255	280	320	8.7	1,125	1,300	1,490	1,600	1,850	16.75
200	230	260	285	330	8.8	1,170	1,355	1,550	1,670	1,920	17.0
210	240	270	290	340	8.9	1,220	1,410	1,610	1,740	1,995	17.25
215	250	280	305	350	9.0*	1,270	1,465	1,670	1,820	2,070	17.5
220	255	285	310	360	9.1	1,320	1,525	1,740	1,900	2,150	17.75
225	260	295	320	370	9.2	1,370	1,585	1,810	1,980	2,230	18.0*
230	265	300	330	380	9.3	1,420	1,645	1,880	2,060	2,315	18.25
240	275	310	340	390	9.4	1,480	1,705	1,950	2,140	2,400	18.5
245	280	320	350	400	9.5	1,540	1,765	2,020	2,220	2,500	18.75
250	290	330	360	415	9.6	1,600	1,830	2,090	2,300	2,600	19.0
260	300	340	370	430	9.7	1,660	1,900	2,160	2,380	2,700	19.25
270	310	350	380	440	9.8	1,720	1,970	2,240	2,460	2,800	19.5
275	320	360	390	450	9.9	1,780	2,040	2,320	2,550	2,900	19.75
280	325	370	400	460	10.0*	1,850	2,120	2,410	2,650	3,000	20.0*
290	335	380	410	475	10.1	1,930	2,210	2,520	2,800	3,200	20.5
295	345	390	420	485	10.2	2,020	2,420	2,650	2,960	3,400	21.0
305	350	400	435	500	10.3	2,120	2,540	2,800	3,130	3,620	21.5
310	360	410	445	510	10.4	2,230	2,680	2,960	3,310	3,840	22.0*
320	370	420	455	525	10.5	2,350	2,840	3,130	3,500	4,100	22.5
330	380	430	470	540	10.6	2,490	3,010	3,310	3,700	4,330	23.0
335	390	445	480	555	10.7	2,640	3,190	3,500	3,920	4,600	23.5
345	400	455	495	570	10.8	2,800	3,380	3,700	4,160	4,880	24.0*
350	410	470	505	585	10.9	2,970	3,580	3,920	4,420	5,200	24.5
360	420	480	520	600	11.0	3,150	3,790	4,160	4,710	5,420	25.0
370	430	490	530	615	11.1	3,340	4,010	4,420	5,030	......	25.5
380	440	500	545	630	11.2	3,540	4,240	4,710	5,380	......	26.0*
390	450	515	555	640	11.3	3,750	4,480	5,030	......	......	26.5
400	460	525	570	655	11.4	3,970	4,730	5,390	......	......	27.0
410	470	535	580	670	11.5	4,200	4,990	......	......	......	27.5
420	485	550	595	685	11.6	4,440	5,270	......	......	......	28.0*
430	495	560	610	700	11.7	4,690	5,580	......	......	......	28.5
440	510	575	620	720	11.8	4,950	......	......	......	......	29.0
450	520	585	635	735	11.9	5,220	......	......	......	......	29.5
						5,500	......	......	......	......	30.0*

*Indicates Commercial Sizes.

Table 57. Conversion Table of Round Pipe Diameters to Wall-Stack Sizes (Standard*)

Round Pipe Diam.	Wall Stack Size	Round Pipe Diam.	Wall Stack Size	Round Pipe Diam.	Wall Stack Size
5.2	8×3	6.3	*12×3	8.9	14×5
5.7	8×3½	6.9	*12×3½	9.5	14×5½
5.5	*9×3	7.4	12×4	6.9	15×3
6.0	9×3½	7.9	12×4½	7.6	15×3½
5.8	*10×3	8.3	12×5	8.2	15×4
6.3	*10×3½	8.8	12×5½	8.7	15×4½
6.8	10×4	6.5	13×3	9.2	15×5
7.2	10×4½	7.1	13×3½	9.8	15×5½
7.7	10×5	7.7	13×4	7.1	16×3
8.1	10×5½	8.2	13×4½	7.8	16×3½
6.0	11×3	8.7	13×5	8.4	16×4
6.6	11×3½	9.2	13×5½	9.0	16×4
7.1	11×4	6.7	*14×3	9.0	16×4½
7.6	11×4½	7.4	*14×3½	9.5	16×5
8.0	11×5	7.9	14×4	10.1	16×5½
8.5	11×5½	8.5	14×4½		

Table 58. Conversion Table of Round Pipe Diameters to Rectangular Ducts 7, 8, 9, 10, and 12″ Deep

Round Pipe Diameters in Inches Equivalent to the Following Rectangular Sizes					Width of Duct
7″ Deep	8″ Deep	9″ Deep	10″ Deep	12″ Deep	
5.0	5.2	5.5	5.8	6.3	3
5.8	6.1	6.5	6.8	7.4	4
6.5	6.9	7.3	7.7	8.3	5
7.2	7.6	8.0	8.4	9.2	6
7.7	8.2	8.7	9.2	10.0	7
8.2	8.8	9.3	9.8	10.7	8
8.7	9.3	9.9	10.4	11.4	9
9.2	9.8	10.4	11.0	12.0	10
9.6	10.2	10.9	11.5	12.6	11
10.0	10.7	11.4	12.0	13.2	12
10.4	11.1	11.8	12.5	13.7	13
10.8	11.5	12.3	12.9	14.3	14
11.1	11.9	12.7	13.4	14.7	15
11.4	12.3	13.1	13.8	15.2	16
11.8	12.6	13.5	14.2	15.7	17
12.1	13.0	13.8	14.6	16.1	18
12.4	13.3	14.2	15.0	16.5	19
12.7	13.6	14.5	15.4	17.0	20
12.9	13.9	14.9	15.7	17.4	21
13.2	14.2	15.2	16.1	17.8	22
13.5	14.5	15.5	16.4	18.1	23
13.8	14.8	15.8	16.8	18.5	24
14.0	15.1	16.1	17.0	18.8	25
14.3	15.4	16.4	17.3	19.2	26
14.5	15.6	16.7	17.6	19.5	27
14.8	15.9	17.0	18.0	19.8	28
....	16.1	17.2	18.2	20.2	29
....	16.4	17.5	18.5	20.5	30
....	16.6	17.7	18.8	20.8	31
....	16.9	18.0	19.1	21.1	32
....	....	18.2	19.3	21.4	33
....	....	18.5	19.6	21.6	34
....	....	18.7	19.8	21.9	35
....	....	19.0	20.1	22.2	36
....	....	....	20.3	22.5	37
....	....	....	20.6	22.8	38
....	....	....	20.8	23.0	39
....	....	....	21.1	23.3	40
....	....	....	....	23.5	41
....	....	....	....	23.8	42
....	....	....	....	24.0	43
....	....	....	....	24.3	44
....	....	....	....	24.5	45
....	....	....	....	24.8	46
....	....	....	....	25.0	47
....	....	....	....	25.2	48

Section 6. *Insulation.* Insulate all exposed warm-air ducts in attic space or under unexcavated unheated sections, with not less than two layers of air-cell asbestos paper or equal. In cold attic spaces more insulation is desirable.

Section 7. *Duct Supports.* All ducts shall be securely suspended from adjacent building members.

Section 8. *Volume Dampers.* Volume dampers of locking type must be placed in each warm-air run, from 6 to 12 inches from the main trunk. Splitter dampers may be necessary at any branch. Return-air ducts *should* be similarly equipped and outside-air inlet *must* be similarly equipped.

Table 59. Equivalent Length of Round and Rectangular Elbows, Angles and Other Turns

To convert the dynamic loss in a *round elbow* to an *equivalent* length of pipe in feet, multiply diameter of elbow in inches by *number* of *diameters* offering equal *resistance* from Table 59A and divide by 12 inches. Add equivalent lengths of all elbows and turns to the *actual* length of straight pipe in feet to obtain the total equivalent length from the *furnace* to the register.

Table 59A

Radius of inside throat in per cent of rd. pipe diameter...	0	¼	½	¾	1	1¼	1½
Diameters of round pipe offering equal resistance.........	38	18.5	13	10	8.5	8	7.5

To convert the dynamic loss in a *rectangular* elbow to *equivalent length of duct in feet,* multiply *width of elbow* in inches on the side making the turn by number of widths offering equal resistance, (see Table 59B) and divide by 12 inches. Add equivalent lengths of all elbows and turns to actual length of straight pipe in feet to obtain the *total equivalent length* from *furnace* to register.

Table 59B

Radius of inside throat in per cent of pipe width.........	0	¼	½	¾	1	1¼	1½
Widths of rectangular pipe offering equal resistance......	47.5	16.5	9	6	5	4.5	4

45 degree angles offer approximately half the resistance of 90 degree elbows. Side wall and baseboard register boxes offer approximately half the resistance of a square throat elbow.

Table 60. Correction Tables for Pipes of Unequal Equivalent Lengths

Equivalent Length of Round Pipe in Feet	Correction Factor	Equivalent Length of Round Pipe in Feet	Correction Factor	Equivalent Length of Round Pipe in Feet	Correction Factor
200	1.150	130	1.050	60	.905
190	1.140	120	1.035	50	.87
180	1.125	110	1.020	40	.835
170	1.115	100	1.000	30	.785
160	1.090	90	.985	20	.725
150	1.075	80	.965	10	.63
140	1.065	70	.93	...	....

The above table is for the purpose of correcting the diameter of round pipes of unequal equivalent lengths, in order that any or all pipes, regardless of their equivalent lengths, will handle any required c.f.m. at the same predetermined static pressure.

Table 61. Properties of Dry Air

Temp. Deg. F.	Density or Weight of Air in Lbs. per Cu. Ft.	% of Volume at 70 Deg. F.	Temp. Deg. F.	Density or Weight of Air in Lbs. per Cu. Ft.	% of Volume at 70 Deg. F.	Temp. Deg. F.	Density or Weight of Air in Lbs. per Cu. Ft.	% of Volume at 70 Deg. F.
Zero	.08636	.868	75	.07124	1.0095	140	.0662	1.132
10	.08453	.8867	110	.06968	1.0756	145	.06565	1.1417
20	.08276	.9057	115	.06908	1.0850	150	.0651	1.1512
40	.07945	.9434	120	.06848	1.0945	155	.0645	1.1620
60	.0764	.9811	125	.0679	1.104	160	.06406	1.1700
65	.07567	.9905	130	.06732	1.1133	...	......	......
70	.07495	1.000	135	.06675	1.123	...	......	......

Practical Application of the Technical Code. The Code has been so designed as to apply to all average conditions for small structures. The methods of procedure as outlined are *standard* methods *suggested* for the calculation of heat losses, furnace sizes, duct sizes, and the general apparatus for heating and air-conditioning systems. These methods aim to simplify the engineering work necessary in applying heating and air-conditioning principles to given cases. As explained in the Code, these methods were constructed for the most part by averaging the field experiences over a long period of time which covered a large number of typical cases.

The Code is easy to follow in doing the necessary engineering work on common type residences and other small structures. It is assumed that the reader will be able to solve any average heating or air-conditioning problem if he will carefully parallel the outline starting on page 114 and calculate heat losses by using such formulas as given in the Code. Therefore, no illustration of the exact Code method is given here.

In industry many engineers use somewhat different methods to work out heat losses and all the other calculations and designs for a heating and air-conditioning system. However, all such different methods are very close to the Code method, especially in terms of final results. For example, in Section 8 the Technical Code gives directions for calculating U values and heat losses whereas in Chapter V, Vol. II, Formulas (1) and (6) do exactly the same things. The methods in Chapter V, Vol. II, give results equally as correct as those suggested in the Code. In fact here, as in many other places, the Code methods were derived from methods in Chapter V, Vol. II. The main points to be brought out here are first, the Code method is absolutely correct; second, it can be used by

readers unfamiliar with other methods; third, other methods can be used equally as well; and fourth, a combination of Code and other methods can also be used.

As a typical example of another method, the following example is given. In Chapter XIII a method combining the Code and other methods is presented for a typical structure under ordinary conditions.

EXAMPLE

Figs. 71 to 74 show the working drawings for a typical frame residence. It is required to design a winter air-conditioning system and to show plans for duct layout and register (warm and cold air) locations. Assume gas as the fuel.

Note: In the following solution of the above example, many details are omitted because of identical previous explanations. Such places will be noted throughout the solution.

***Solution.** The heat losses should be calculated first. No U value calculations are given here because it is assumed that if the reader knew the construction and insulation specifications he could easily calculate them by using the methods given in the Technical Code. Therefore no specifications are given here and U values are assumed.

Note: Dimensions, U values, etc., in the solution are approximate following the practical methods employed by the average engineer. For example, dimensions are taken in feet and tenths of feet and U values may be taken as 1.1 in place of 1.13, etc.

The heat losses for the various rooms are as follows:

**Bedroom No. 1	=13×13.7×7.5#	
Cubic Content	=1335	
Glass	=22 square feet× 1.1	=24.2
Outside Wall	=188 square feet×.131	=24.628
Ceiling	=178 square feet×.123	=21.894
††Infiltration	=1335×.02	=26.7
	Total	=97.422
†97.422×70°	=6,819.54 B.t.u. loss.	
Bedroom No. 2	=11×12×7.5	
Cubic content	=990	
Glass	= 18 square feet× 1.1	=19.8
Outside wall	=177 square feet×.131	=23.187
Ceiling	=132 square feet×.123	=16.236
Infiltration	=990×.02	=19.8
	Total	=79.023
79.023×70°	=5,531.61 B.t.u. loss.	

* Data Courtesy of The Henry Furnace & Foundry Co.—Cleveland, Ohio.

** This method is a variation of Formula (I) in Chapter V, Vol. II and Formulas (29) to (38) in Chapter VI.

† Temperature difference = 70° F.

†† Another method of calculating infiltration.

\# Dimensions are approximate. Therefore the reader may not obtain exactly the same results for balance of rooms. This is usual procedure. Where room shapes are irregular the dimensions can be averaged as for Den.

Bath	$=12.2\times6.7\times7.5$
Cubic content	$=615$
Glass	$= 8$ square feet $\times\ 1.1 = 8.8$
Outside wall	$= 90$ square feet $\times .131 = 11.79$
Ceiling	$= 82$ square feet $\times .123 = 10.086$
Infiltration	$=615\times.02 = 12.3$
	Total $= 42.97$
$42.97\times80°$*	$=3{,}437.6$ B.t.u. loss.
Hall No. 1	$=4\times10.5\times7.5$
Cubic content	$=315$
Glass	$= 5$ square feet $\times\ 1.1 = 5.5$
Outside wall	$= 25$ square feet $\times .131 = 3.275$
Ceiling	$= 42$ square feet $\times .123 = 5.166$
Infiltration	$=315$ square feet $\times\ .02 = 6.3$
	Total $= 20.241$
$20.241\times70°$	$=1{,}416.87$ B.t.u. loss.
Den	$=7.5\times9.5\times7.5$
Cubic content	$=533$
Glass	$= 8$ square feet $\times\ 1.1 = 8.8$
Outside walls	$=131$ square feet $\times .131 = 15.851$
Ceiling	$= 71$ square feet $\times .123 = 8.733$
Infiltration	$=533\times.02 = 10.66$
	Total $= 44.044$

$44.044\times70° = 3{,}083.08$ B.t.u. loss.

In like manner the other rooms are calculated and the B.t.u. losses found to be as follows:

#Living room	=11,521.30 B.t.u.
Dining room	= 5,843.74 B.t.u.
Kitchen	= 4,655.84 B.t.u.
Breakfast room	= 1,701.28 B.t.u.
Hall No. 2	= 2,079.77 B.t.u.
Lavatory	= 761.04 B.t.u.
Recreation room	= 9,267.51 B.t.u.
Garage	=10,067.70 B.t.u.

The total B.t.u. loss for the entire house = 66,186.88.**

The next step is to calculate the C.F.M. This is done by multiplying the B.t.u. loss by .0122. The factor .0122 is found by the following formula.

$$\text{B.t.u.} = \frac{\text{C.F.M. } (T_1 - T_2)\ 60}{55} = \tag{44}$$

$$\text{C.F.M.} = \frac{55 \text{ B.t.u.}}{(140-70)\times60} \quad \frac{55 \text{ B.t.u.}}{70\times60} = .0122 \text{ B.t.u.}$$

In the above formula it is evident that T_1 = register temperature and T_2 = room temperature.

* Bathroom temperature is 10° higher than other rooms.

** It is pointed out again that the solution assumes the reader could determine U values, etc., as explained in other chapters (including Chapter IX) and therefore these calculations are not shown here.

\# Dimensions are approximate. Therefore the reader may not obtain exactly the same results for balance of rooms. This is usual procedure. Where room shapes are irregular the dimensions can be averaged as for Den.

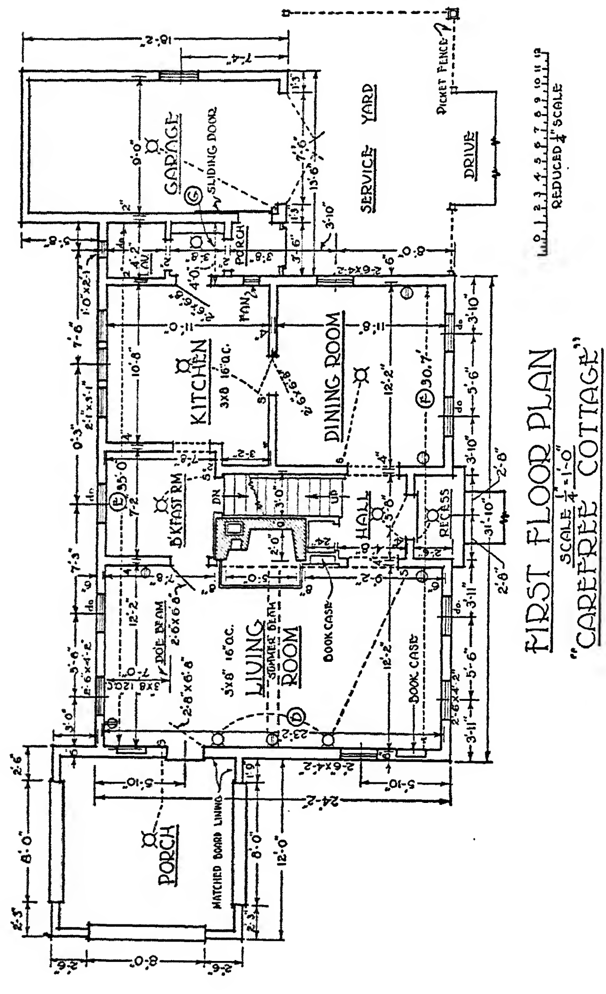


Fig. 71. First Floor Plan—Carefree Cottage
Courtesy of M. R. Johnke, Architect

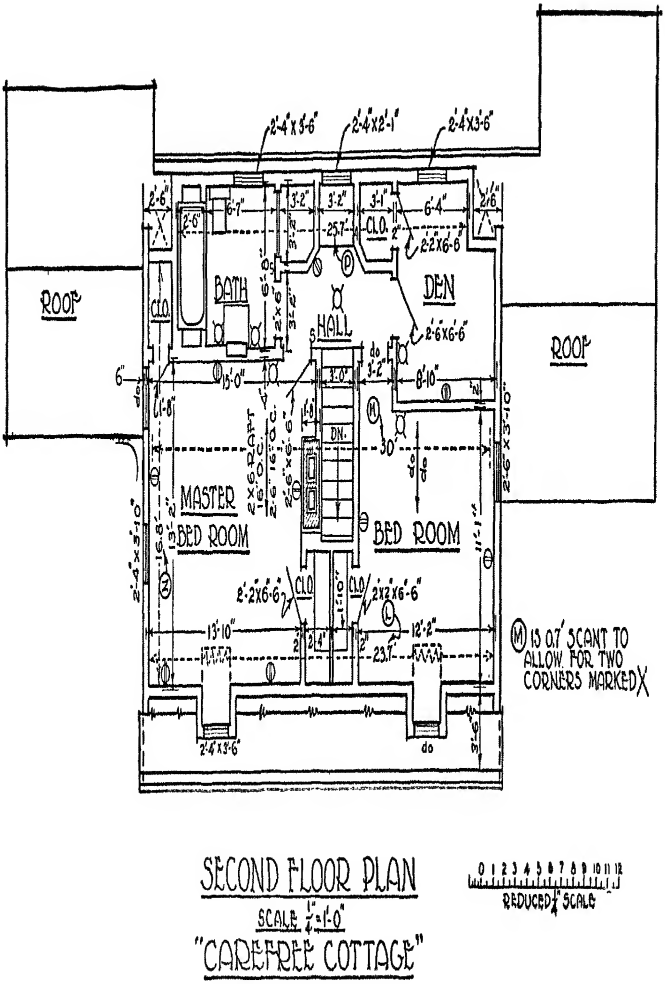
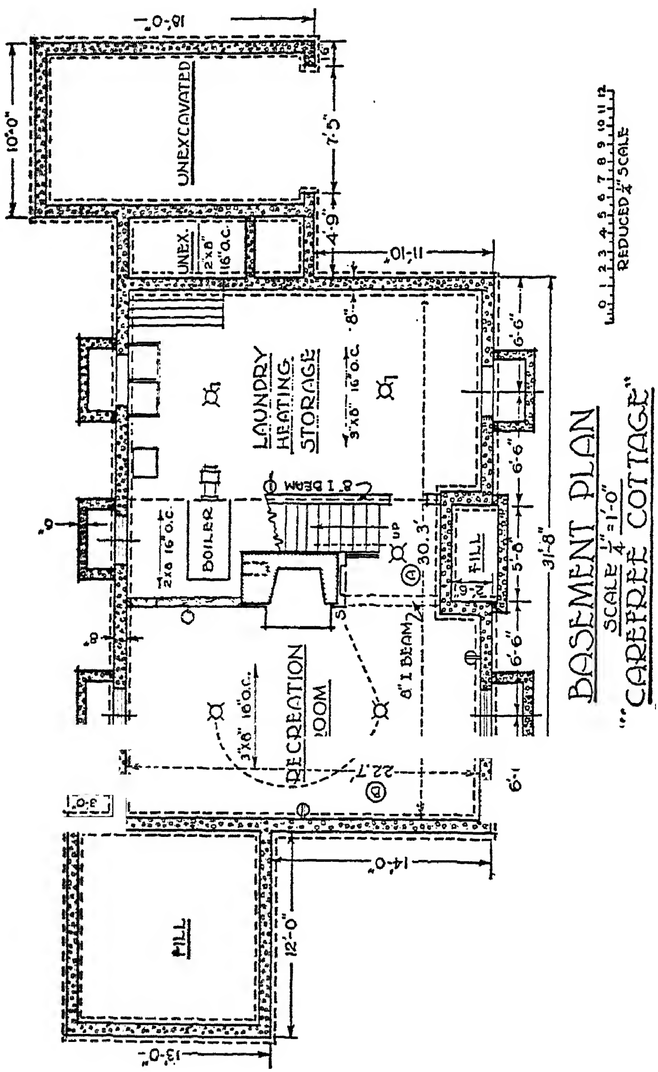


Fig. 72. Second Floor Plan—Carefree Cottage
Courtesy of M. R. Johnke, Architect



75 ...asement Plan— ... Cot...
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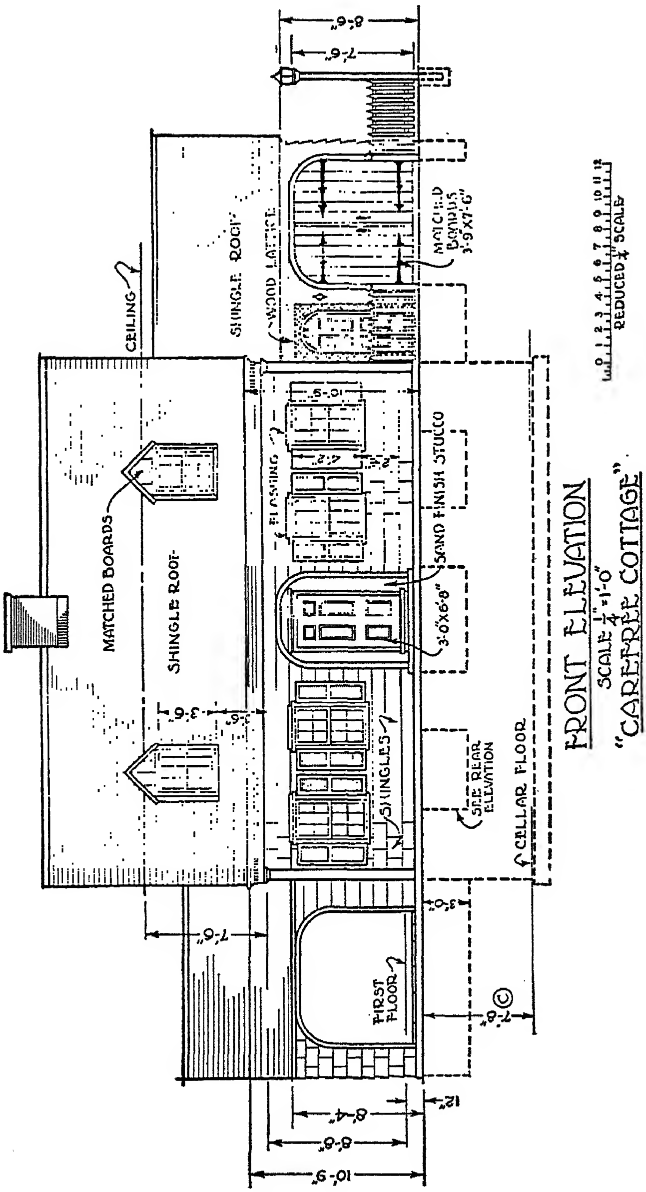


Fig. 74. Front Elevation—Carefree Cottage
Courtesy of M. R. Johnke, Architect

Next the areas of the warm air areas are found by multiplying the C.F.M. by the following factor which can be determined by the following formula.*

$$\frac{V\times A}{144}=\frac{300\times A}{144}=\text{C.F.M.} \qquad (45)$$

$$\frac{144\text{ C.F.M.}}{V}\ \therefore A=\frac{144\text{ C.F.M.}}{300}=.48\times\text{C.F.M.}=A$$

In Formula (45) it is evident that V = velocity and A = Area.

Using the above Formulas (44) and (45) the C.F.M. and warm air areas are found to be as shown in Table 62.

Table 62. Data for Winter Air-Conditioning Example

Room	B.t.u. Loss	C.F.M.	W. A. Area
Bedroom No. 1	6,819.54	82.2	39.4
Bedroom No. 2	5,531.61	67.5	32.4
Bathroom	3,437.60	42.0	20.2
Hall No. 1	1,416.87	17.3	8.4
Den	3,083.08	37.6	18.0
Living room	11,521.30	140.1	67.3
Dining room	5,843.74	71.3	34.2
Kitchen	4,655.84	57.0	27.4
Breakfast room	1,701.28	20.8	10.0
Hall No. 2	2,079.77	25.4	12.4
Lavatory	761.04	9.3	44.6
Recreation room	9,267.51	113.0	54.2
Garage	10,067.70	123.0	59.0

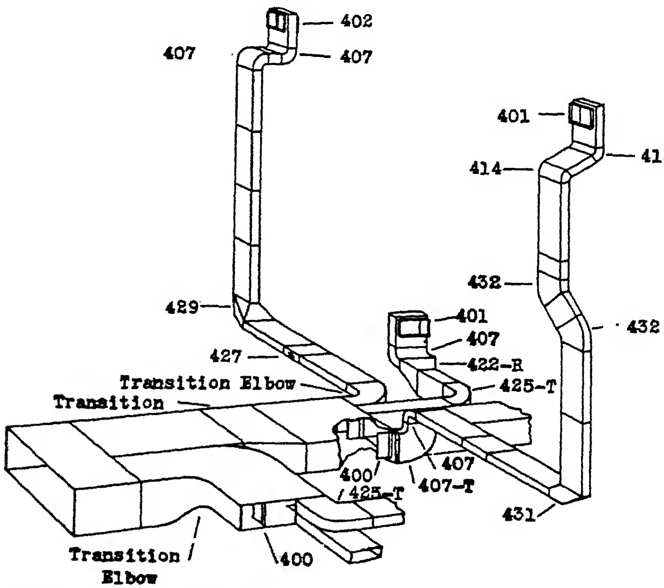


Fig. 75. Perspective of a Typical Duct System Showing Transition Pieces. Fig. 76 Shows Such Pieces in Place in the System for the Example Being Worked Out. The Numbers Shown in Fig. 75 Are Catalogue Numbers.

* See also Chapter XIII for more detailed explanation and refer to the Code.

To determine the register sizes, use standard register sizes, as in Chapter VIII. Knowing the warm air area needed, registers are selected having a free area of the same or slightly greater area. The Tables in Chapter VIII may also be used for this purpose.

The stack sizes are calculated for 400 feet per minute by using the following formula.

$$A = \frac{\text{C.F.M.} \times V}{144} \qquad (46)$$

In most cases stacks must run between studs in walls and partitions so their width cannot be greater than approximately 14 inches at a maximum.

In calculating duct sizes, this method adds 20 per cent to the C.F.M. of the room. Then adding the C.F.M.'s, starting at a point farthest from the furnace, and transitioning (see Figs. 75 and 76) after each group of branches if needed.*

C.F.M.+20%		Branch Duct Size	
68	21.0	3×8	
85	26.2	4×8	
100	30.8	4×8	
253	**78**	11×8	transition
	68	9×8	
100	26.8	4×8	
85	22.8	3×8	
62	16.6	3×8	
500	**134**	19×8	

The bold face figures are taken from a friction chart, Fig. 70, using a constant friction rather than constant velocity, which is .04. Then that area is proportioned between each register. For example:

$$\frac{68}{253} \text{ of } 78 = 21$$

$$\frac{85}{253} \text{ of } 78 = 26.2$$

This is carried on to the end of the duct. Fig. 76 shows the various ducts, their sizes, and locations. Figs. 78 and 79 show locations and sizes of all registers.

The following material explains the location of the items mentioned in the previous paragraph.

Ordinarily the garage and bathroom need not have returns for cold air. The kitchen would come under this classification, too, as the cooking odors might be circulated throughout the residence if return air was taken from this room.

The location of inlets (warm air registers or grilles) should, if possible, be on inside walls. If they are in outside walls the ducts must be insulated all of which adds materially to the cost of the system. On the other hand, as for gravity furnaces, the cold air faces (registers or grilles) should be located on outside walls.

* See Chapter XIII for more detailed explanation and refer also to the Code.

Thus, in most cases, the inlets and outlets may be placed on opposite sides of the rooms to help bring out the required circulation.

The location of the inlets and outlets depends on a number of other items besides those previously given. Naturally the location of an inlet or an outlet determines the position of the stacks (ducts). Therefore in locating an inlet, for example, the plans, Figs. 76 to 79, must be studied to ascertain if a duct can be brought to that point. It must be kept in mind that the duct comes from the air conditioner and must have free access through the walls, etc. It could easily be possible to decide on an inlet location only to find that it would be impossible to bring a duct to that point. Or, maybe bringing a duct to that point could easily involve too many turns and too long a run. The engineer, in designing a duct system, should strive to make the duct runs as short as possible and eliminate as many turns as possible. Thus it can be seen that a most careful study is necessary.

Let us suppose that the air conditioner must be located in the basement, as shown in Fig. 76, where it is out of the way and does not take up any valuable basement space. Then as a preliminary example we will locate the inlet for the den, Fig. 79. There are two inside walls in this room wherein the inlet could be located. If the inlet is located in the wall between the den and bedroom No. 1 the duct would have to run between the floor joists for a distance because there is no first floor partition directly under this second floor partition.

Note: It is suggested that the reader draw all floor plans, to the quarter inch scale, on vellum or tracing cloth and tack them all down on a drafting board so that the second floor is on top, then the first floor, and finally the basement. If all drawings are properly lined up, the outside walls for all three sheets will coincide thus facilitating the location of ducts by being able to see through the tracing cloth to the second floor plan and to the first floor plan, etc.

In such a case the duct would have to be carried down through the partition between the kitchen and dining room at the expense of two 90° turns.

Examination of the other inside wall shows that the short wall space between the two doors could be utilized for an inlet. This is shown at point 7 in Fig. 79. From this point the duct would have to run between the joists, a matter of two or three feet, to a point over the first floor partition between the breakfast room and kitchen. The duct (stack) could run down this partition point 25 in Fig. 78 to a point 48 in Fig. 76 just above the conditioner.

The advantage of the point 7 location over the first one considered is that the latter makes possible a much shorter duct run.

The inlet for bedroom No. 1 was placed at point 10 because of the ease in running the stack from point 10 horizontally between joists to point 11 and then down through the continuous partition. In Fig. 78 this is shown at point 33 and at point 65 in Fig. 76. The other inside wall of bedroom No. 1 cannot be used because there is no supporting or near partition on the first floor. A careful study will show that point 10 is the only convenient and possible short run location for the inlet.

For bedroom No. 2 the inlet location at point 5 was selected so as to have it on the coldest side of the room. The duct, point 16, runs from point 5 horizontally between joists to point 4 where it goes down the partition as noted at point 32 in Fig. 78 and point 53 in Fig. 76.

The inlet for the bathroom is located at point 1 and the duct, point 2, runs horizontally between joists to point 3 where it goes down through the first floor partition, point 31, and to the basement ceiling at point 54.

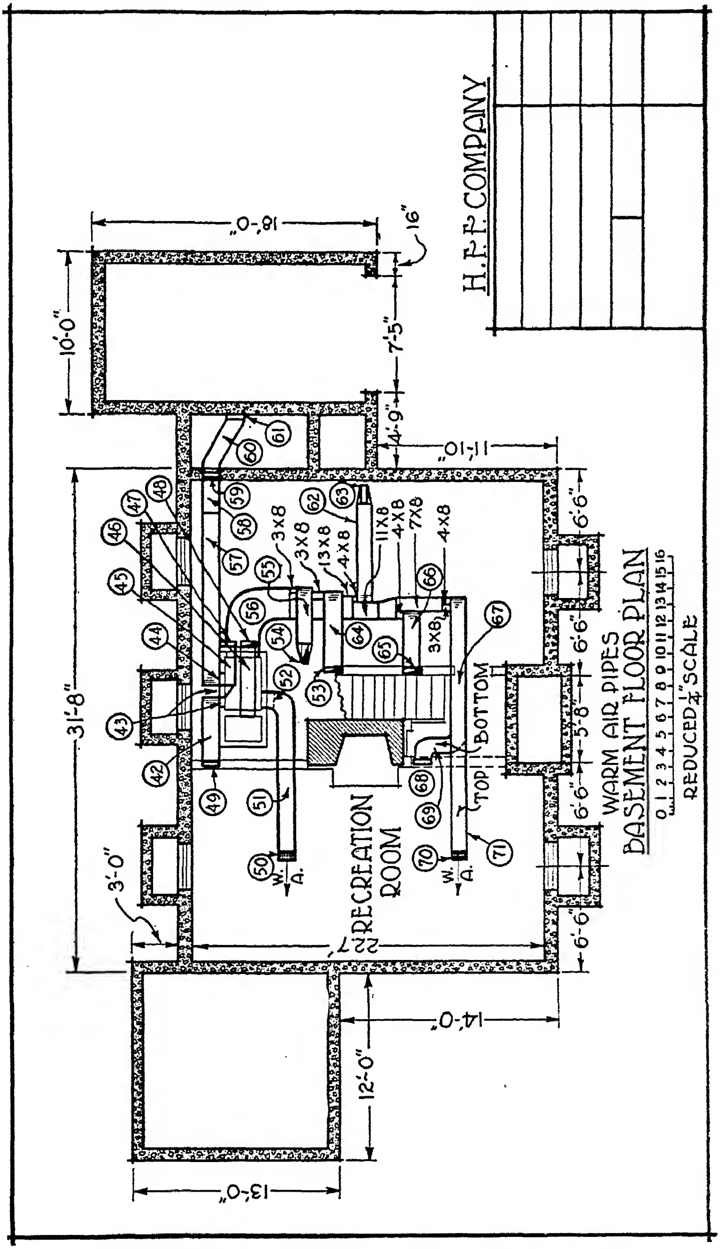


Fig. 76. Basement Floor Plan Showing Warm-Air Ducts

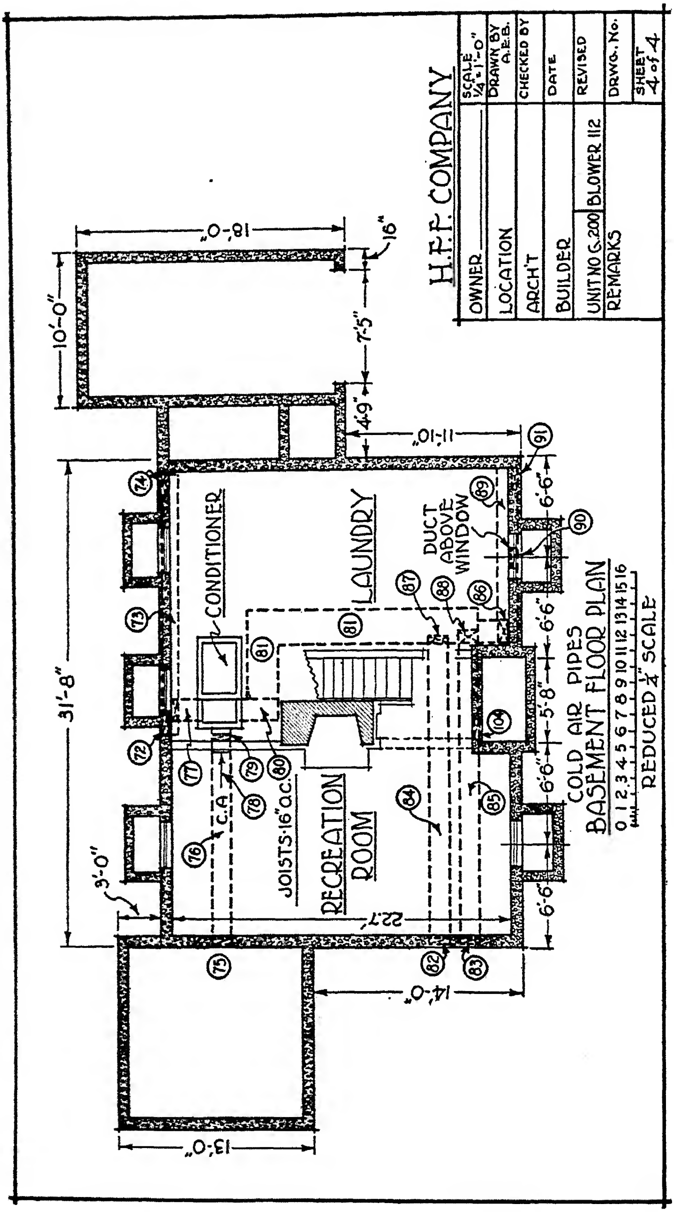


Fig. 77. Basement Floor Plan Showing Cold-Air Ducts

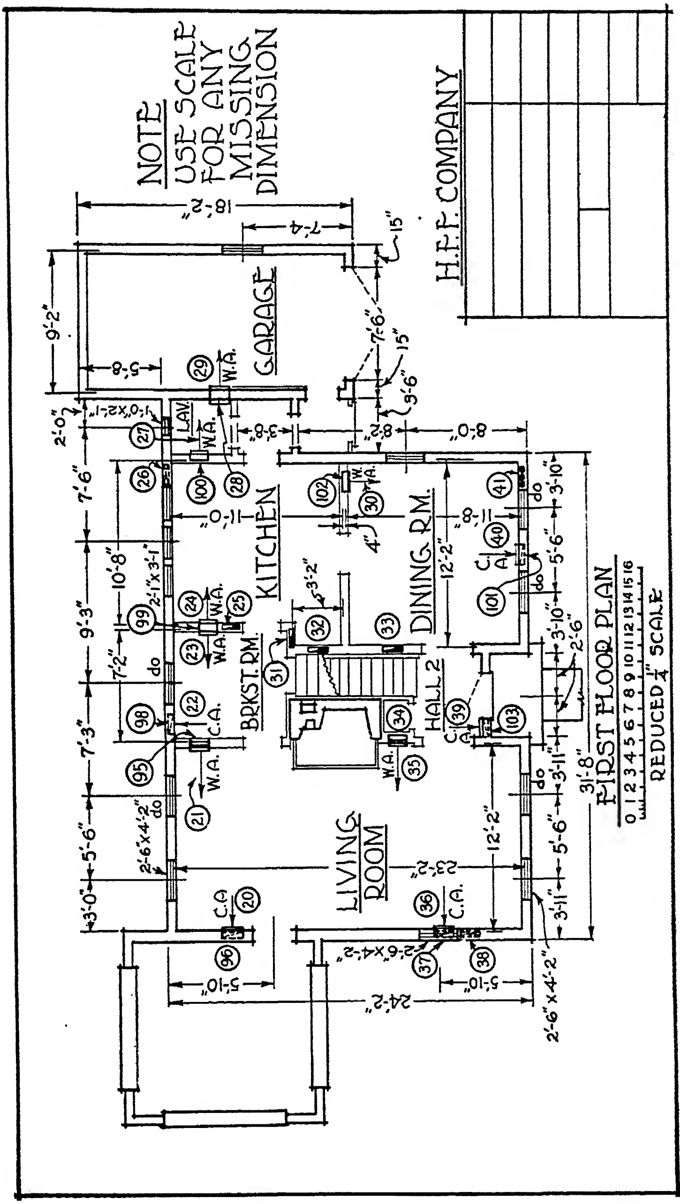


Fig. 78. First Floor Plan Showing Warm- and Cold-Air Registers

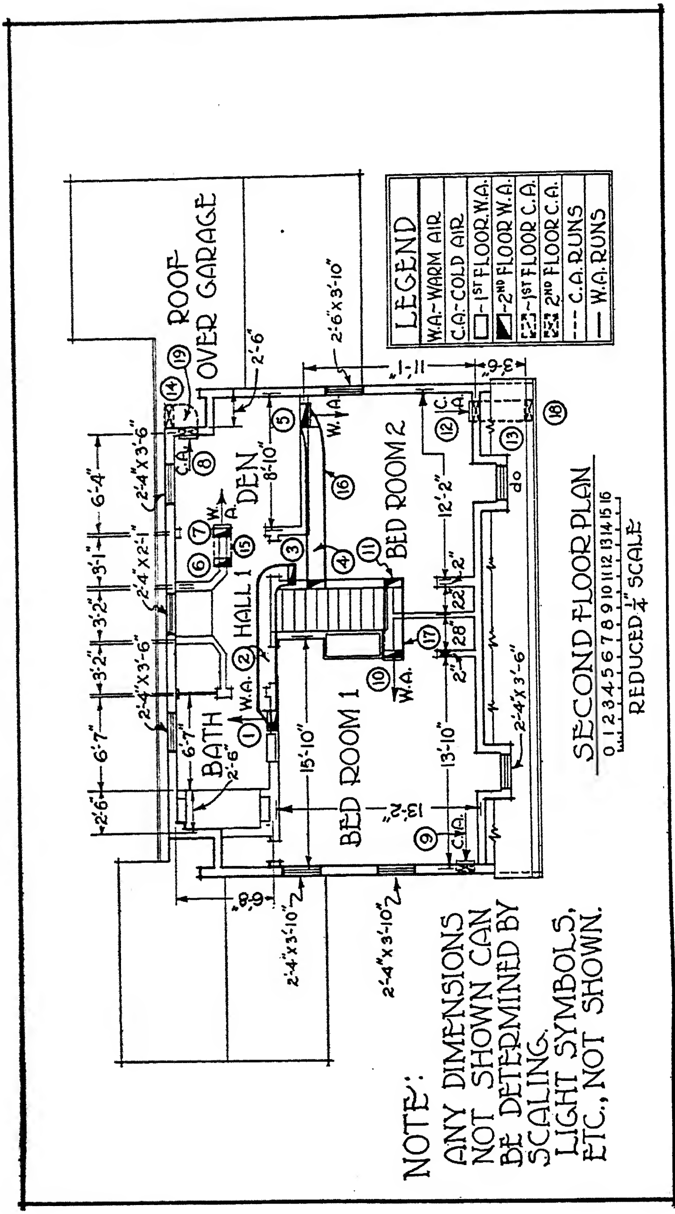


Fig. 79. Second Floor Plan Showing Warm- and Cold-Air Registers

By studying Fig. 76, it can be seen that the second floor stacks emerge into the basement, points 65, 53, 54, and 48, in such a manner as to make possible the use of a trunk duct which starts at point 56. The stack for the den, point 48, is too far from the other second floor stacks to use the trunk duct but it does come close to the conditioner. Thus it can be seen that by some thought and study the stacks can be grouped so as to make possible the use of a trunk duct.

Note 1: The warm and cold air ducts in the basement are shown on separate drawings for clearness. In practice both should be on the same drawing.

Note 2: In actual practice the sizes of all ducts, inlets, outlets, etc., should appear on the drawing. Here a legend system is used because of the reduced size of the drawings.

The inlets and ducts for the first floor rooms were decided on in much the same manner as explained for the second floor.

The inlets for the recreation room were put in central locations convenient to duct runs.

Ordinarily the garage duct is an individual duct, but the lavatory being close by and because of convenience and economy the duct, point 57 in Fig. 76, supplies both inlets at 27 and 29 in Fig. 78.

The locations of cold air outlets and their stacks are usually much simpler to ascertain because of their employing outside walls. The outlet for bedroom No. 2 is at point 9. This is opposite the inlet as recommended. The stack is carried down the outside wall, point 38 in Fig. 78, and into the basement at point 83 in Fig. 77. In locating outlets it is advisible to have them opposite the inlets and at the same time make it possible to carry the basement runs between joists. Sometimes the inlet openings have to be changed on account of meeting necessary requirements in the outlet ducts.

The cold air ducts should enter the basement so as to make a trunk duct return possible, as shown in Fig. 77. This makes less duct work necessary and prevents covering too much of the ceiling with ducts.

There are possibly quite a number of other arrangements for inlets, outlets, ducts, etc., than the design covers in this example. The designer should strive to select or design the system that has the shortest total run, is the most economical, requires the least cutting of framing parts, and makes as little visible duct work in the basement as possible.

In the following description, all of the numbered points are explained and the specifications shown. The reader may draw Figs. 76 to 79 to the ¼-inch scale and insert the specifications at the proper places should the following method prove difficult to understand or use in his study.

DESCRIPTION OF NUMBERS SHOWN IN FIGS. 76 TO 79

Den (Fig. 79)

7. Warm air register. Size 12×6 inches.
8. Cold air register. Size 14×6 inches.
6. Warm air stack.
14. Cold air stack.
15. Horizontal duct running between joists. Size 12×3½ inches.
19. Horizontal duct. Size 14×4 inches.

Bathroom (Fig. 79)

1. Warm air register. Size 12×6 inches.
3. Warm air stack.
2. Horizontal duct running between joists. Size 12×3½ inches.

Bedroom No. 2 (Fig. 79)

5. Warm air register. Size 12 × 6 inches.
12. Cold air register. Size 14 × 6 inches.
4. Warm air stack.
18. Cold air stack.
16. Horizontal duct. Size 12 × 3½ inches.
13. Horizontal duct. Size 14 × 4 inches.

Bedroom No. 1 (Fig. 79)

10. Warm air register. Size 12 × 8 inches.
9. Cold air register. Size 14 × 6 inches.
11. Warm air stack.
17. Horizontal duct running between joists. Size 12 × 3½ inches.

Living Room (Fig. 78)

21 and 35. Warm air registers. Size 12 × 6 inches.
20 and 36. Cold air registers. Size 14 × 6 inches.
95 and 34. Warm air ducts.
96 and 37. Cold air ducts.
38. Cold stack from bedroom No. 1.

Breakfast Room (Fig. 78)

23. Warm air register. Size 12 × 6 inches.
22. Cold air register. Size 14 × 6 inches.
99. Warm air duct.
98. Cold air duct.

Kitchen (Fig. 78)

24. Warm air register. Size 12 × 6 inches.
99. Warm air duct.
26. Cold air duct from den.
31. Warm air duct going to bathroom.
32. Warm air duct going to bedroom No. 2.
25. Warm air duct going to den.

Lavatory (Fig. 78)

27. Warm air register. Size 12 × 6 inches.
100. Warm air duct.

Garage (Fig. 78)

29. Warm air register. Size 10 × 12 inches.
28. Warm air duct.

Dining Room (Fig. 78)

30. Warm air register. Size 12 × 6 inches.
40. Cold air register. Size 14 × 6 inches.
102. Warm air duct.
101. Cold air duct.
41. Cold air duct from bedroom No. 2.
33. Warm air duct going to bedroom No. 1.

Hall No. 2 (Fig. 78)

39. Cold air register. Size 14 × 6 inches.
103. Cold air duct.

Basement (Fig. 76)

70 and 50. Warm air registers in recreation room. Sizes 12 × 6 inches.
51 and 71. Warm air ducts running between joists. Sizes 12 × 3½ inches.
67. This part of the warm duct supplies both recreation and living room registers. It runs between joists. Size 12 × 7 inches.
69. This is a branch from duct 67. Size 12 × 3½ inches.
68. Warm air duct to living room register.
65. Warm air duct to bedroom No. 1.
66. Warm air duct. Size 12 × 3½. Between joists.
53. Warm air duct to bedroom No. 2. Size 12 × 3½ inches. Between joists.
64. Warm air duct. Size 12 × 3½ inches. Between joists.
54. Warm air duct to bathroom. Size 12 × 3½ inches. Between joists.
55. Warm air duct. Size 12 × 3½ inches. Between joists.
62. Warm air duct. Size 12 × 3½ inches. Between joists.
63. Warm air duct going to dining room.
61. Warm air duct to garage.
69. Warm air duct. Size 12 × 3½ inches.
59. Warm air duct going to lavatory.
58. Duct divides into two parts here. One part goes to lavatory and one to garage.

57. Warm air duct. Size 12×9 inches. Between joists.
56. Warm air trunk duct. Size 19×8 inches.
46. Warm air duct. Size 12×3½ inches. Between joists.
48. Warm air duct going to den.
47. Warm air duct going to kitchen and breakfast room.
45. Warm air duct. Size 12×3½ inches. Between joists.
44. 9×8 inches.
43. 7×6 inches.
42. Warm air duct. Size 12×3½ inches. Between joists.
49. Warm air duct going to living room.
52. Warm air rise into joists.

Basement (Fig. 77)

82. Cold air duct from bedroom No. 2.
83. Cold air duct from living room.
75. Cold air duct from living room.
84. Cold air duct between joists.
85. Cold air duct between joists.
76. Cold air duct between joists.
78. Cold air register for recreation room. Size 14×6 inches.
79. Riser—14×4 inches.
77. Cold air duct. Size 14×7 inches.
72. Cold air duct from breakfast room.
80. Cold air duct. Size 18×12 inches.
81. Cold air duct. Size 24×8 inches.
73. Cold air duct between joists.
74. Cold air stack from den.
87. Opening 14×4 inches.
88. Opening 14×7 inches.
89. Cold air duct between joists.
90. Cold air stack from dining room.
91. Cold air stack from bedroom No. 2.
86. Size 14×7 inches.
104. Cold air stack from hall No. 2.

Cold air returns are figured at 250 C.F.M. in this example, at register or outlet, and ducts are figured on constant friction, which is .04 on friction chart.

Note: The example in Chapter XIII gives more detailed explanations relative to the design of cold air returns.

Table 62A. Moncrief Gas Furnaces for Air-Conditioning Systems

Furnace Number	Number of Burners	Dia. Flue Pipe	Size Gas Line	Heating Surface	Casing Free Area	Velocity Through Furnace	B.t.u. Input	Forced Air B.t.u. at Register	C.F.M.	Blower Size	Motor Hp.	Number of Filters	Velocity Through Filters
50	1	3	⅜	2,160	233	...	30,000	......	.....	...	..	.	...
100	2	4	1	4,320	376	215	60,000	42,750	560	110	¼	2	105
150	3	5	1	6,480	438	277	90,000	64,125	840	110	¼	2	158
200	4	6	1	8,640	584	277	120,000	85,500	1,120	112	¼	3	140
250	5	7	1¼	10,800	730	277	150,000	106,875	1,400	112	¼	4	131
300	6	8	1¼	12,960	976	248	180,000	128,250	1,680	114	⅓	4	158
350	7	8	1¼	15,120	1,146	246	210,000	149,625	1,960	114	⅓	5	147
400	8	9	1¼	17,280	1,168	277	240,000	171,000	2,240	212	½	5	168
450	9	10	1¼	19,440	1,314	276	270,000	192,250	2,520	212	½	5	183
500	10	10	1¼	21,600	1,376	293	300,000	213,750	2,800	214	½	6	175
600	12	10	1½	25,920	1,678	288	360,000	256,500	3,360	214	¾	6	210

The selection of the furnace is a simple task because the Code specifies the use of manufacturer's ratings for gas- or oil-fired apparatus.

Table 62A shows a typical data sheet put out by one manufacturer.

Knowing the total heat loss to be 66,186+B.t.u. per hour, it is a simple matter to select the number 200 furnace as the most logical size. This particular furnace gives ample factor of safety for extremely cold days and takes care of design temperatures easily.

The blower is an integral part of the furnace system, as shown in Fig. 80, as is also the humidifier. All such apparatus is designed to the correct size and capac-

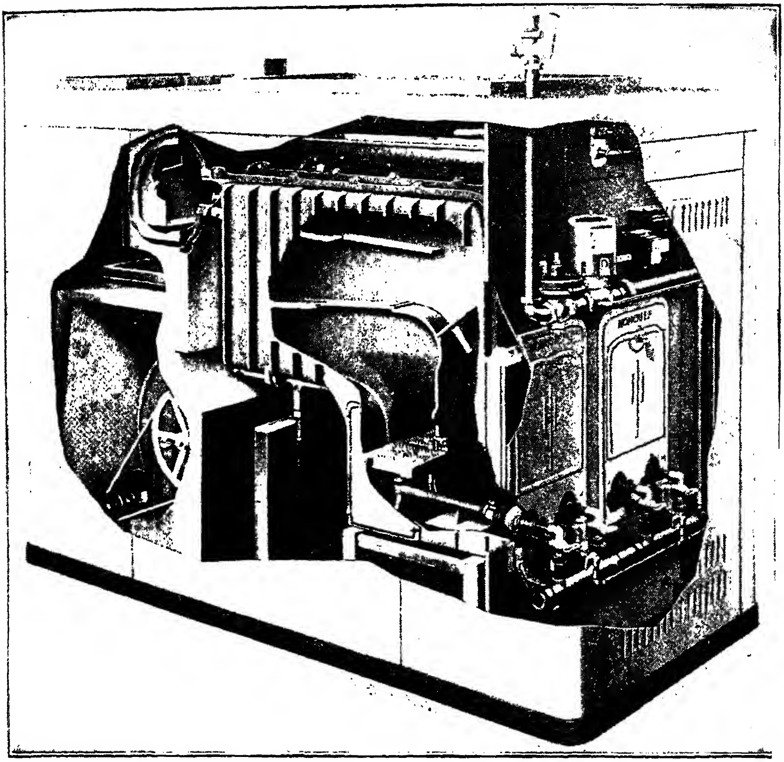


Fig. 80. Moncrief Gas-Fired Furnace Having Blower, Filters, and Humidifiers as Integral Parts of the Furnace
Courtesy of the Henry Furnace and Foundry Company

ity to fit the needs of each particular furnace. Therefore once the furnace has been selected no further selection of such apparatus is necessary. The controls can be fully automatic or manual as is fully described in Chapter XII.

PRACTICE PROBLEMS

Figs. 81 and 82 are required to work out the following problems. The specifications are as follows:

Basement: Concrete foundation 12 inches thick. Floor 4 inches thick. Basement temperature can be assumed 20°F. lower than the temperature of living rooms.

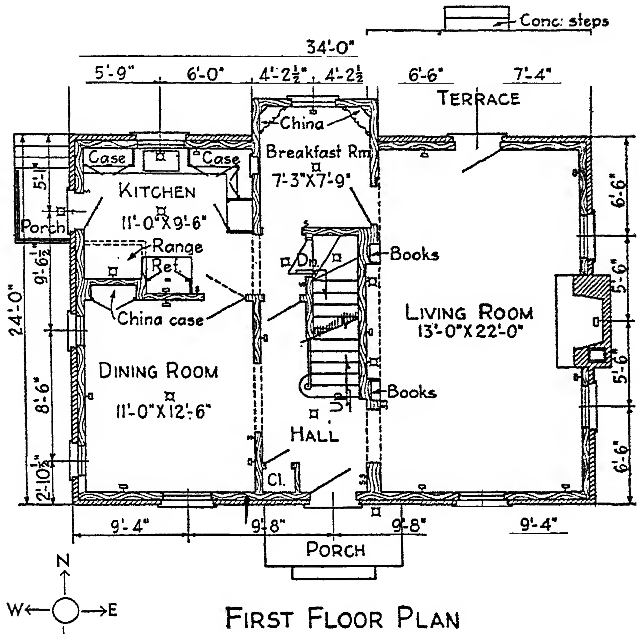


Fig. 81. First Floor Plan

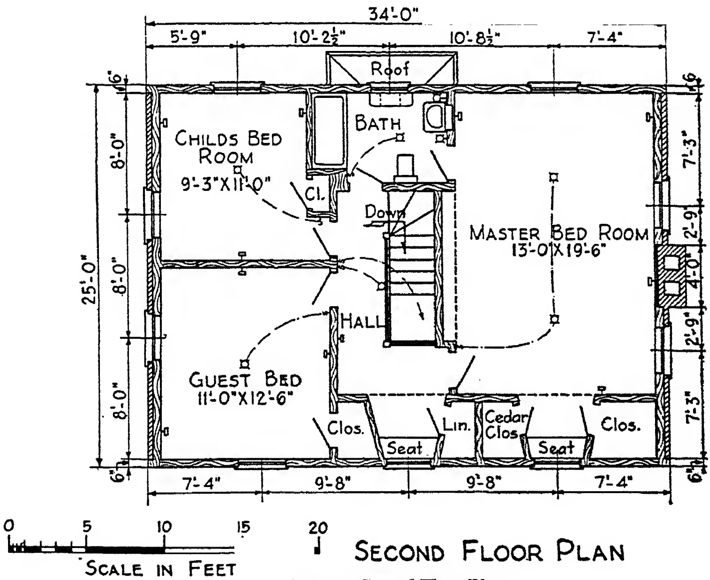


Fig. 82. Second Floor Plan

Side Walls: Brick veneer. Sheathing (⅞-inch Insulite) is nailed to 2x4 studs. 4-inch brick veneer is directly against the sheathing. Lath is ½-inch Lok-Joint. Plaster is ½-inch thick.

First Floor: Joists are 2x10 spaced 16 inches apart. Rough flooring is pine ⅞-inch thick. Finish flooring is oak ⅜-inch thick. ½-inch Insulite insulation is nailed to underside of joists.

Attic Floor: Joists are 2x8 spaced 16 inches apart. Rough flooring is ⅞-inch pine. Wood lath and plaster form the second floor ceiling. 4 inches of Eagle Wool is placed in between flooring and lath.

Roof: Rafters are 2x6 spaced 16 inches apart. Roof boards are close fitting ⅝-inch thick. Asbestos shingles are used. The vertex of the roof is 9 feet above attic floor. Attic windows are tight and well constructed. Attic temperature can be assumed as half the difference between inside and outside temperatures.

Fireplace Chimney: Consider two courses of brick.

Doors: All exterior doors 3′–0″ x 7′–0″. Interior doors 2′–6″ x 7′–0″. Doors are of the usual thickness and of wood.

Windows: All windows are double hung. Assume window height 5½ feet. Narrow dining room windows are 2 feet wide. All other windows are 3 feet wide. Kitchen window 4 feet high. All windows are of wood sash, good construction, and weatherstripped.

Temperature: Assume minimum outside temperature is −16°F. The inside temperature is 70°F. Wind is generally N.W. at 15 m.p.h. Breathing line temperatures need not be considered.

1. Assume all rooms, including upstairs and downstairs halls, are to be heated. The attic and basement are not heated. Use the Technical Code and calculate the total heat loss per hour for the house.

2. Using the Technical Code, calculate the required grate area for a hand-fired coal furnace. The grate area to the heating surface ratio equals 20–1.

3. Explain what full-winter air conditioning is and what it accomplishes. Specify such a system for the residence in Problems 1 and 2.

4. Make drawings of the floor plans of the residence of Problems 1 and 2, showing locations of furnace, registers or grilles, and all rectangular ducts as shown in Figs. 76, 77, 78, and 79. Also draw any special details necessary to show any irregular duct runs as shown in Figs. 51, 52, and 53. After Problem 5 has been completed, the correct dimensions can be put on these drawings.

5. Using the Technical Code method, calculate the proper sizes for all ducts and registers or grilles.

6. Suppose the residence referred to in Problems 1 and 2 had an old style round gravity furnace. How would you proceed in redesigning the system to make it mechanical and air conditioned? Draw a sketch of your design. Explain the operation.

7. How would you determine the amount of filters needed in the residence referred to in Problems 1 and 2?

8. From the mechanical furnaces (air conditioning for winter only) shown in the text, select one that you feel will amply fill the need of the residence being considered. Explain what system of winter air conditioning you prefer and tell why.

Note: Problems 9 through 12 are to be solved after studying Chapter VI.

9. Assume that the living room, in Fig. 81, is to be cooled during the sum

mer when outside temperature is 95°F. dry bulb and 75°F. wet bulb to an inside temperature condition of 80°F. dry bulb and 65°F. wet bulb. No other part of the house is to be cooled. Assume that basement temperature is 88°F. and the master bedroom, Fig. 82, 95°F. Do not consider any infiltration through window cracks nor the inside doors which can be assumed closed at all times. The greatest number of occupants is 12 and each must have 10 cubic feet of air per minute. There are 500 watts in lighting. The south window has a canvas awning. Calculate the cooling and dehumidifying load in tons of refrigeration.

10. Suppose a system, as shown in Fig. 92, was being applied to the residence. If the living room, at the cooling and dehumidifying load calculated in Problem 9, and the dining room, requiring half the load of the living room, were to be cooled during the day to the temperatures given in Problem 9, design the cooling coil assuming such coils as the ones shown in Fig. 91.

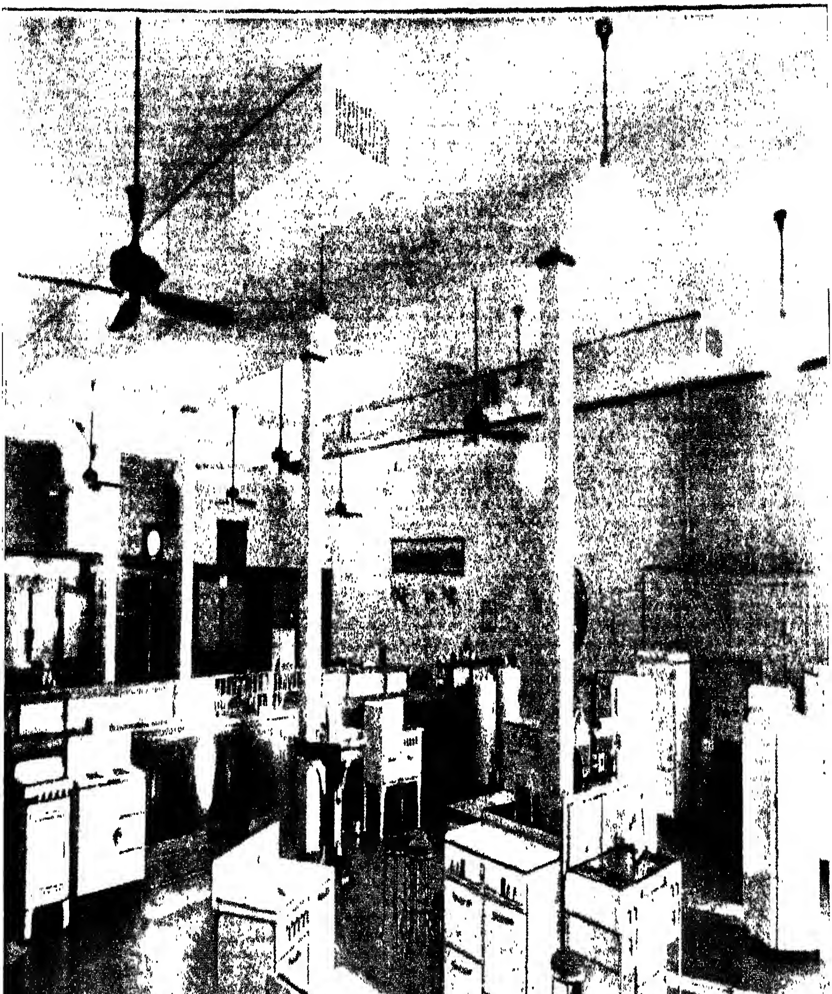
11. Assume a locality where day-time temperatures of 97°F. are common and where night-time temperatures fall to 77°F. Explain how you would take advantage of such a condition to make night and day cooling of a residence more economical.

12. Assuming the temperatures given in Problem 11, how would you take advantage of the conditions to cool a residence by natural means without the aid of cooling apparatus, such as coils or air washers.

13. Is recirculation of air in a residential mechanical system advisable? Give reasons to prove your answer. Assume air conditioning.

14. Is the use of about 75 per cent outside air in a residential mechanical air-conditioning system economical? Give reasons to prove your answer for both winter and summer operation of the system.

15. Is some form of mechanical or automatic fuel feeder advisable in mechanical residential air-conditioning systems? Give reasons to prove your answer.



DUCTS IN TYPICAL DISPLAY ROOM

Courtesy of Fairbanks-Morse Company

CHAPTER VII

AIR-CONDITIONING FURNACES

A mechanical furnace may be without any air-conditioning apparatus, in which case it has only a blower as shown in Fig. 57 or with filters, like in Fig. 64. Such a furnace might easily have a form of a humidifier such as Fig. 126 or Fig. 127 in the section on "Humidification" and would then function only to heat, humidify, and circulate the air in winter, and provide some measure of cooling in the summer by circulating the air throughout the house. During the evening and night hours when the outdoor air cools down a few degrees, the air may be taken from the outside and produce cooling as is explained later in this chapter. This type of mechanical furnace is not recommended except where extreme economy is required.

It is recommended that all mechanical furnaces have some form of air cleaners and means for humidifying the air. In some furnaces air cleaning is accomplished by filters (Figs. 59, 62, and 63) and in others it is done by air washers (Fig. 58). In still other types the air cleaning is done by both filters and air washers (Fig. 83).

Contrary to general belief, all mechanical furnaces are not equipped to condition the air the year round. Furnaces can be grouped as follows.

Group 1. Winter conditioning.

Group 2. Winter and summer conditioning.

Figs. 62 and 63 would come under Group 1 because the equipment shown is entirely for winter conditioning. The air is filtered, heated, humidified, and circulated. No provision is made for summer cooling or dehumidifying.

Fig. 58 would come under Group 2 because the unit performs all winter functions together with the added summer functions of cooling and dehumidifying. It is understood that some means must be provided of supplying cold water to the air washer for air cooling. This is explained later in this section.

Such winter conditioners as shown in Figs. 62 and 63 can be made into summer conditioners by simply adding cooling coils to the

outlet ducts. This would also require a refrigeration machine unless cold water was available to circulate through the coils in place of a refrigerant.*

Fig. 83 shows a diagram† of a mechanical air-conditioning furnace in which all major parts are named. In this typical system the air returns from the rooms to *A* via the return ducts. Or, some recirculated and some outdoor (fresh) air may be used, in which case

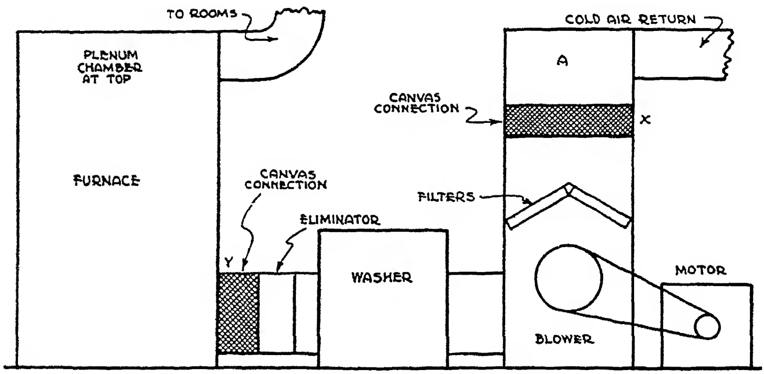


Fig. 83. Air-Conditioning System Using Filters and Washers

both recirculated and fresh air are mixed at *A*. In such a case, a fresh-air duct would be needed besides the recirculating (cold-air) ducts. Sometimes all fresh air is used, in which case no recirculating ducts are required. However, air is generally recirculated because the recirculated air does not require much heating, making the system more economical in operation. From *A* the air is drawn through the filters, then forced through the air washer where it is washed and humidified before entering the furnace where it is heated. The plenum chamber is an enclosure where the air is kept under pressure so as to feed out into all leaders in equal amounts, or at the same rate per square inch of leader. A system of this kind could be operated by using either a filter or an air washer.

The system in Fig. 83 could be converted into a summer cooling, dehumidifying, and cleaning system either by supplying cold or refrigerated water to the air washer or installing a cooling coil just

*See page 180 where a typical illustrative example of cooling coil calculations is given.

†This diagram illustrates the process of air conditioning in mechanical furnaces such as shown in Fig. 58. The apparatus is different but the principle is the same.

below the filters. Such an installation would require a refrigerating machine in addition to the blower.

There are a great many makes of air-conditioning furnaces but all of them operate on the same general principles outlined for Fig. 83.

Converting Old Furnaces. Most old furnaces are of the round or square types, as shown in Figs. 40, 41, 42, and 43. Such furnaces were designed without any thought of air conditioning other than possibly a slight humidifying effect. It is doubtful if it would be economically sound to convert such types into mechanical air-conditioning furnaces but in some cases it might prove desirable.

Fig. 60 illustrates what might be accomplished in converting an old gravity warm-air furnace into an air-conditioning mechanical furnace. If it is assumed that the round furnace is an old one, then the conversion means the addition of the filters and blower, all of which are shown in the apparatus to the right of the furnace. These blowers and filters are selected, depending on the capacity or requirements of the residence, for example, from manufacturers' catalogues and ratings.

In Fig. 60 the return-air ducts all meet above the blower if recirculating systems are used. In the system illustrated in Figs. 51, 52, and 53, the cold-air ducts end 18 inches above the floor, and the cold air, or return air, is taken directly from the basement.

The blower and filter unit discharges air into the furnace casing and from that point on, the system operates as in a gravity furnace, except that the circulation is under pressure.

Mechanical furnaces have different leader-area requirements than gravity furnaces, as will be seen by comparing the Gravity and Technical Codes. Thus some changes in the leaders might be necessary in an old system being converted.

Fig. 83 also might be used as an illustration of converting an old gravity furnace if the furnace shown is assumed as being, for example, a round gravity type. The old return or recirculating ducts, as shown in Fig. 40, would be removed and their openings in the furnace casing sealed. The apparatus in Fig. 83 could then be installed and used as already described. The cold-air returns would be near the ceiling and all meet at *A*.

If leaders are not replaced by new and redesigned leaders, the distribution of air to the various rooms will be uneven. In other words, first-floor rooms near the furnace will receive too much air whereas second-floor rooms may not receive enough. To remedy this, dampers can be installed in each leader and kept partially shut or open full, depending on results of actual trials.

Automatic firing of coal, gas, or oil can be incorporated in a converted or new system. Fig. 60 shows an automatic stoker. The

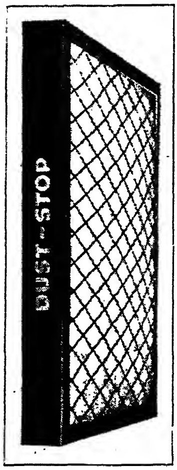


Fig. 84. Typical Filter
Courtesy of Owens-Illinois Glass Company

stoker and blower can be controlled by thermostats located in the rooms and full automatic control obtained.

In a converted system a new humidifier will be needed unless a pan-type humidifier is already in place in the bonnet. (See section on "Humidification".)

Fig. 83 shows canvas connections at *X* and *Y*. These connections are the means of insulating the residence from the blower so that noises and vibration cannot be transmitted to the rooms via the leaders or cold-air return pipes. All mechanical furnaces, whether new or converted, should be insulated in this manner when the blower is apart from the main furnace or source of heat.

Furnace Air-Conditioning Apparatus. The air-conditioning apparatus used with mechanical or gravity furnaces is generally

incorporated as an integral part of a complete unit as shown in Fig. 58, and are bought as one unit. The selection is, then, a matter of consulting manufacturers' catalogues and selecting a unit capable of supplying the required heat, cooling, etc. In other cases, units are built to supply only what is called winter conditioning and contain blowers, filters or washers, or both, humidifiers, and automatic controls. If summer cooling beyond minimum results is desired, cooling coils or refrigerated water can be added. In this case the coils must be selected by manufacturers' catalogue ratings. The same thing is true if an air washer is added.

The most general method of positive cooling in residences, is by means of coils and a compressor. Coils can be inserted into such a system as shown in Fig. 83, if no air washer is used between the filters and blower. The compressor would be in a remote place with pipes to and from the coil. In the system shown in Fig. 64, the coils could be placed in the part marked X. The unit-type air conditioner such as shown in Fig. 58 is perhaps most satisfactory, because all apparatus is compactly placed within one small casing.

Filters. Fig. 84 shows a typical filter such as is generally used in furnace work. This is called the "throw-away" type because, once the filter has become saturated with dirt, it can be discarded and replaced economically by a new one. Fig. 85 shows a type of filter that is more expensive but which can be cleaned with oil. The cleaning is done with an odorless oil of a viscosity, such as Gulf No. 1561. A tank of sufficient depth to submerge the filter is used. To clean a filter, work it up and down in the cleaning oil (in tank). Invert and repeat the operation several times. Small particles, such as lint can be brushed off the inlet side with a whisk broom. Allow the filter to drain at least 10 minutes. A filter of this kind works on the impingement principle, which means that it is charged with clean oil. After cleaning, the filter is dipped into another oil bath and is again ready for service. The cleaning oil can be used repeatedly if the dirt is removed after settlement. This type of filter should be cleaned often enough to insure against its causing undue resistance to the passage of air.

For gravity furnaces, such as shown in Fig. 45, the filters, typical types like those shown in Fig. 84, are selected as shown by the following example.

Example. A gravity warm-air installation having four warm-air runs or leaders is as follows:

1 leader, 8 inches—area	50 square inches
1 leader, 9 inches—area	63 square inches
1 leader, 10 inches—area	78 square inches
1 leader, 12 inches—area	113 square inches
Total pipe area,	304 square inches

For illustrative purposes, the 16×25×2-inch filter will be used.

$304 \times 5 = 1{,}520$ square inches

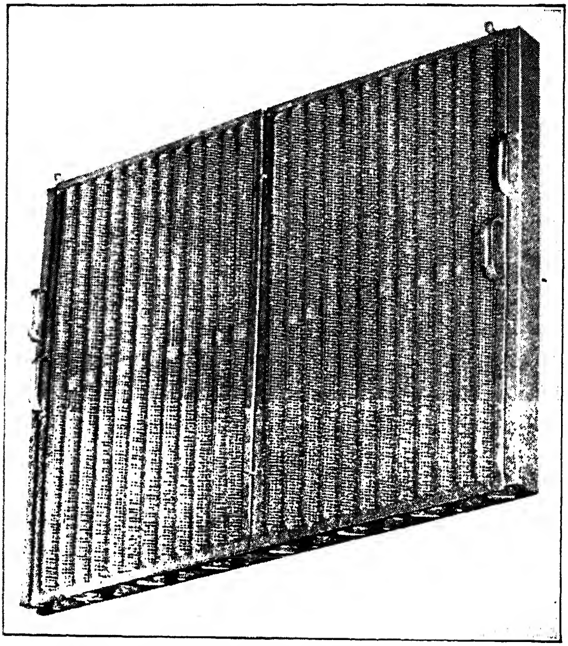


Fig. 85. Typical Filter Which Can Be Cleansed with Oil
Courtesy of Schwitzer-Cummins Co., Indianapolis, Indiana

The area of the 16×25×2-inch filter is 400 square inches. Then $1{,}520 \div 400 = 3.8$ filters required. The fraction is considered as an additional filter so 4 filters are required.

For mechanical furnaces, the selection of filters should be carried on according to the ratings supplied by manufacturers in their catalogues. Air filters used for air conditioning are purchased by the number of cubic feet of air delivered by the fan. For example, if a fan delivers 1,200 c.f.m., the filter must also deliver 1,200 c.f.m., plus approximately 10 per cent for safety, or 1,320 c.f.m.

Air Washers. Air washers used with residence or other small systems may be an integral part of the apparatus, as shown in Fig.

58, or as a separate piece of apparatus as shown in Fig. 83. There are many good washers available and manufacturers' catalogues can be obtained showing them.

An air washer operates so that water, under a pressure, is forced through spray nozzles which send the water out in a fine mist. The air is caused to go through this mist and, in doing so, takes up

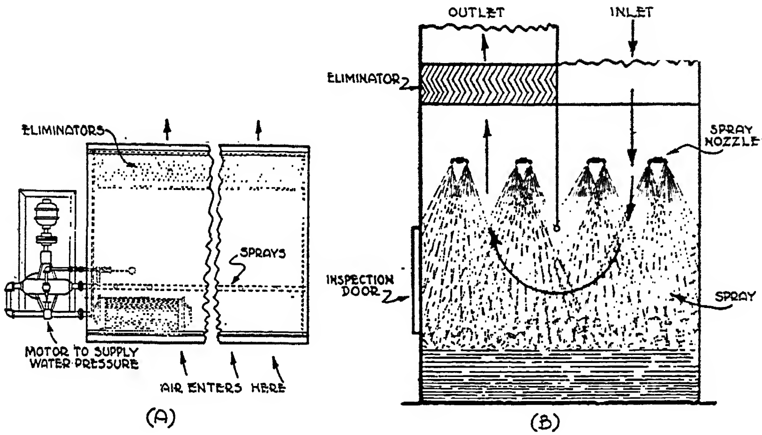


Fig. 86. Two Typical Air Washers

moisture or loses moisture, depending on water temperature, and becomes clean. In its cleaning effect the washer takes out all particles of dust, pollen, etc.

In the winter an air washer functions to clean and humidify the air. The air is forced through the apparatus and, as explained, must pass through a dense mist where much of the dirt, etc., is wet and falls to the bottom of the washer. The air in leaving the spray chamber must pass through eliminators which are composed of a series of metal plates so arranged that the air zig-zags or changes its direction many times. This serves to take out all entrained water and also aids in taking dirt out of the air. Because of their weight, particles of dirt cannot quickly change their direction when being carried by air through eliminators, with the result that they bump against the eliminator plates and, because these plates are wet, the particles stick and are washed down to the bottom of the washer.

Fig. 86 shows a plan and a vertical section view of two typical air washers. Most washers are controlled by solenoid valves wired

in parallel with the fan motor. The water supply may be controlled by a controlling device in the rooms. In the summer the washer may function to clean, dehumidify, and cool the air. The air is cleaned, as in winter operation.

The size of washers depends first on the available space. However, for residences and other small enclosures, the washer size need

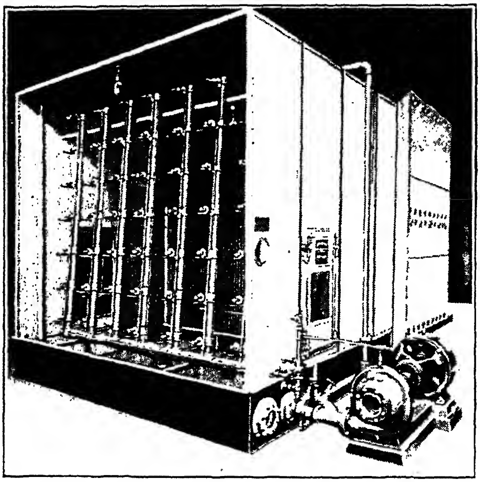


Fig. 87. Typical Air Washer for Large Installations
Courtesy of The American Blower Company

not be large. For large installations, such as public buildings, washers of the type shown in Fig. 87 are used. Such washers are usually rated in catalogues in terms of cubic feet of air per minute, which ratings are based on an air velocity of 500 feet per minute. Typical sizes run from 4½ feet to 9 feet in length, depending on whether or not cooling coils are installed in the washer, the number of spray banks, and the thicknesses of the entering and the leaving eliminators. The area of a washer is often affected by space restrictions or by space specifications on the inside of a washer. Such specifications have to do with face areas of cooling coils. Manufacturers' catalogues should be consulted for specific ratings of standard washers as well as for resistance figures.

Air washers are generally constructed of sheet metal. Doors are put at convenient locations for inspection and cleaning.

If air passes through a washer in which the water circulates without the addition or subtraction of heat, it becomes (approxi-

mately) saturated at its entering wet-bulb temperature. In such a case, the total heat content of the air does not change because the dry-bulb temperature drops in proportion to the additional water evaporated. This is approximately complete adiabatic saturation.

Note: When dry air is combined with water vapor, the temperature of the water is lowered below the temperature of the original dry air. After this process has taken place, the air is the same temperature as the water. A heat transfer of this kind is called adiabatic saturation of the air. This process in an air washer is called evaporative cooling and depends, in efficiency, on the number of spray banks in the washer.

If the air going through a washer comes in contact with water that is heated or cooled, a heat exchange between water and air takes place and the air becomes (approximately) saturated at the temperature of the water as it leaves. If heated water is furnished a washer, the air is warmed and humidified, and the heat given up by the water equals the heat taken up by the air. If cooled water is used in the washer, it cools the air and dehumidification takes place, and the heat taken up by the water equals the heat given up by the air.

The heating or cooling of spray water is accomplished, generally, at a remote point. Steam can be used for heating, as can many standard appliances. Compressors are generally used to cool the water. One means consists of a series of pipes over which the water passes from an overhead distributor. The water is cooled by the evaporation of a refrigerant. Such a cooler is known as a *Baudelot* cooler and is used extensively in air washers for central systems. Cold well water can also be used if available in ample amounts. If well water is used, it passes through the washer once and then to the sewer. Where artificial cooling is done, the water is recirculated and cooled repeatedly by the refrigerating apparatus.

A very common method of cooling spray water is to place the cooling coils in the pan or bottom of the spray chamber. Thus the spray water as it falls to the bottom of the spray chamber is cooled before being pumped through the spray nozzles again.

Air Washer Calculations. The reader is urged to become familiar with the explanation of the uses of the Psychrometric Chart, as given in Chapter IV before going ahead with the problems which follow in this chapter.

The solution of typical air-washer applications, such as Examples 1 and 2 to follow, requires a thorough knowledge of air-conditioning

principles and the reader is again reminded that not much progress can be made until the basic principles of the section on "Air-Conditioning Principles" are thoroughly understood.

Examples 1 and 2, which follow, are typical applications in every detail. No consideration of heat losses or gains is given, as it is assumed that this subject is well understood at this point. The air quantities given are those assumed as required for the two structures mentioned.

Example 1. A residence has six rooms with a bathroom, basement, and attic. There are six occupants in the residence. Air conditioning is to be applied to the interior of the house for dehumidification and cooling during the summer months. The air for the air conditioning is obtained by using 75 per cent recirculated air and 25 per cent outside or fresh air. The residence will require 400 pounds of air per minute. The outside air temperature is 90°F. and it has a relative humidity of 50 per cent. The recirculated air is 80°F. and 50 per cent relative humidity. The interior of the residence is to be maintained at 80°F. and 50 per cent relative humidity. The occupants will provide 700 grains of moisture vapor per person per hour. There are 1,000 grains of moisture vapor per hour mixed with the air of the interior from various service applications, such as equipment of various kinds. An air washer is to be used to dehumidify and cool the air for the residence.

1A. What is the temperature of the air mixture from the exterior and interior entering the air-conditioning system?

1B. What is the quantity of moisture vapor, expressed in grains per pound of air, in the mixture of the air from the exterior and interior?

1C. What is the quantity of moisture vapor, expressed in grains per pound of air, that must be removed from the air for distribution?

1D. What is the quantity of heat, or the heat transfer of the air and water of the washer, to dehumidify and cool the air?

1E. What are the weight and the inlet and discharge temperatures of the water used in the air washer for dehumidification and cooling of the air?

Solution 1A. To determine the temperature of the mixture of recirculated (interior) and exterior air, the following formula can be used.

$$T^M = \frac{T^E \times P^E + T^1 \times P^1}{100} \qquad (47)$$

where T^M = temperature of the mixture of inside and outside air.

T^E = temperature of exterior air.

P^E = percentage of total air supply that is exterior air.

T^1 = temperature of interior air.

P^1 = percentage of total air supply that is interior air.

We can substitute the actual temperatures and percentages in Formula (47) as given in the statement of the example.

Substituting

$$T^M = \frac{90 \times 25 + 80 \times 75}{100}$$

$$T^M = 82.50°\text{F.}$$

Thus, the answer to Example 1A is 82.5°F.

Note: Fig. 88 shows an abbreviated Psychrometric Chart on which are shown lines tracing grains per pound and other figures used in balance of example.

Solution 1B. To determine the amount of moisture vapor, as required in this part of the example, the following formula can be used.

$$G^P = \frac{W^E \times G^E + W^1 \times G^1}{W^E + W^1} \qquad (48)$$

where G^P = the grains of moisture vapor in the mixture of interior and exterior air per pound of air.

W^E = the pounds of dry air from the exterior.

G^E = the grains per pound of dry air from the exterior.

W^1 = the pounds of dry air from the interior or recirculated air.

G^1 = the grains per pound of dry air from interior or recirculated air.

The air at 90°F. and in a saturated condition has 217.6 grains of moisture vapor. (See Fig. 88.) To determine the grains of moisture vapor at the relative humidity

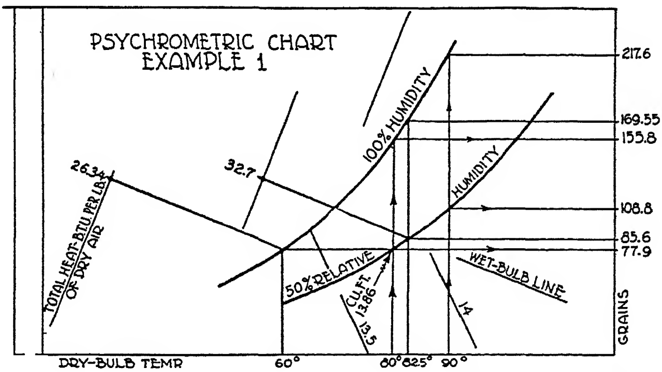


Fig. 88. Abbreviated Psychrometric Chart for Use with Example 1, page 156

Note: The above chart is exactly the same in general principle as the complete chart shown in the back of the book. The methods of using the Abbreviated Chart are the same as explained on page 36. The lines and points shown are those used in the solutions of Example 1, page 156.

of 50 per cent, multiply 217.6 by 50, which equals 108.8 grains. This can also be determined directly from Fig. 88 by following a line from dry-bulb temperature of 90°F. upward until it meets the curved line of 50 per cent relative humidity and from that point of intersection horizontally to the right-hand scale.

Note: The table, chart, and calculation methods are being explained all together so the reader may see the connection between them.

The air from the exterior is 25 per cent of the total supply. To determine the moisture vapor of that quantity of exterior air, multiply 108.8 by 25 per cent. This equals 27.2 grains.

The air at 80°F. in a saturated condition has 155.8 grains of moisture per pound. (See also the chart.) This can also be determined from Fig. 88.

Follow the vertical line from dry-bulb temperature of 80°F. upward until it intersects the 100 per cent humidity curved line. From this point of intersection, follow a horizontal line to scale at the right-hand side of the chart. The 80°F. air is, according to the problem statement, at 50 per cent relative humidity. Therefore to find the grains per pound of air at 80°F. and 50 per cent relative humidity, multiply 155.8 by .50, which gives 77.9 grains. (This can also be found as shown in Fig. 88.) The air from the interior is 75 per cent of the total air supply. Therefore, the moisture vapor content of the amount of interior air is $77.9\times.75=58.4$ grains. Thus the total moisture vapor per pound for the mixture of inside and outside air is $27.2+58.4=85.6$ grains.

Now the example can be worked by using Formula (48).
Substituting in the formula

$$G^P=\frac{100\times 108.8+300\times 77.9}{100+300}$$

$$G^P=85.6 \text{ grains}$$

The 100 and 300 are determined by taking 25 and 75 per cent of 400 pounds of air per minute. This amount is specified in the statement of the example. The 108.8 and 77.9 were calculated in the first part of this solution. It will be noted that this part of the example can be solved either by the simple reasoning method, or by the formula, and that either method requires the use of the chart or Fig. 88. Taking values from the chart is naturally not as accurate as taking them from the table but in many calculations the chart values are sufficiently accurate.

Solution 1C. This example can be solved completely and entirely by the following formula, keeping in mind that data from the previous example, which was solved by the use of the chart and Fig. 80, must be used.

$$G^R=(G^E+G^1)-G^A \tag{49}$$

where G^R = the total grains of moisture vapor to be removed from the air, by the dehumidification process, per hour.

G^E = total grains of moisture vapor from exterior.

G^1 = total grains of moisture vapor from interior.

G^A = total grains of moisture vapor that is mixed with air entering the interior spaces from the air washer after dehumidification has taken place.

Before substituting in Formula (49) it is necessary to calculate G^E, G^1, and G^A.

To calculate G^1 it must first be remembered that only 75 per cent of the interior air is recirculated or sent back to the air washer for dehumidification. The residence requires 400 pounds of air per minute or 400×60 minutes $=24{,}000$ pounds per hour. Of this amount $24{,}000\times.75=18{,}000$ pounds are recirculated from the interior. Air at 80°F. and 50 per cent relative humidity contains 77.9 grains of moisture vapor per pound. Then 18,000 pounds contain $18{,}000\times 77.9$ $=1{,}402{,}200$ grains. To this interior air moisture content, must be added moisture from occupants and service. There are 6 occupants or $700\times 6=4{,}200$ grains per hour. The service conditions add 1000 grains per hour, as given in the example statement. Thus

$1{,}402{,}200+4{,}200+1000=1{,}407{,}400$ grains per hour, value of G^1.

To calculate G^E it must be remembered that only 25 per cent of outside air is used. That means that 25 per cent of 24,000 pounds per hour or 6,000 pounds

is the amount of outside or exterior air supplied per hour. The exterior air has a temperature of 90°F. and a relative humidity of 50 per cent, at which condition it contains 108.8 grains of moisture vapor per pound. (See Fig. 88.)

Then $108.8 \times 6{,}000 = 652{,}800$ grains per hour, value of G^E.

To calculate G^A it must be remembered that the interior air is maintained at 80°F. and 50 per cent relative humidity. There are 24,000 pounds of air per hour at this condition or,

$24{,}000 \times 77.9 = 1{,}869{,}600$ grains per hour, value of G^A.

Therefore $G^1 = 1{,}407{,}400$ grains per hour

$G^E = 652{,}800$ grains per hour

$G^A = 1{,}869{,}600$ grains per hour

Substituting in Formula (49)

$$G^R = 652{,}800 + 1{,}407{,}400 - 1{,}869{,}600$$
$$G^R = 2{,}060{,}200 - 1{,}869{,}600$$
$$G^R = 190{,}600 \text{ grains per hour}$$

The air washer therefore is required to remove 190,600 grains of moisture vapor per hour from the air for distribution to the interior of the residence.

Solution 1D. To determine the heat transfer from the air to the water of the air washer for cooling and dehumidifying, Fig. 88 and standard formulas are used.

The air mixture that is entering the air washer is at a temperature of 82.5°F. (Example 1A). This air contains 85.6 grains of moisture vapor per pound of air (Example 1B). To calculate the heat content, the relative humidity must first be found. Air at 82.5°F. and saturated, contains 169.55 grains of moisture vapor per pound. (See Psychrometric Chart in back of book. In the table this figure is obtained by interpolation between 82° and 83°F.) The relative humidity is found by dividing 85.6 by 169.55 which equals .505 per cent. This is the relative humidity of the air mixture that is entering the air washer from the exterior and interior.

The above calculations are practically the same as applying the following standard formula.

$$R = \frac{G^P}{G^S} \tag{50}$$

where R = the relative humidity of the air and moisture vapor mixture expressed in per cent.

G^P = the grains contained in a partial saturation of the air and moisture vapor mixture per pound.

G^S = the grains contained in a saturated mixture of air and moisture vapor per pound.

Substituting

$$R = \frac{85.6}{169.55}$$

$R = .505$ relative humidity

The chart shows that the heat content of 82.5°F. air is 19.92 B.t.u. This figure is obtained by interpolation. The chart shows the latent heat to be 25.34 B.t.u. To determine the total heat for the air with .505 relative humidity, multiply the 25.34 by the .505 relative humidity which equals 12.80 B.t.u., plus

19.92 B.t.u. sensible heat equals 32.7 B.t.u. which is the heat content of the air at 82.5°F. and .505 relative humidity. This is also shown on the Psychrometric Chart, Fig. 88.

The 190,600 (Example 1C) grains of moisture vapor is divided by 24,000 (pounds of air per hour required by residence) which equals 7.9416 grains. This is the number of grains of moisture vapor that are removed per pound of air by the water in the washer. As the air from the interior and exterior, for the residence, is at 82.5°F. and .505 relative humidity, it has 85.6 grains of moisture vapor per pound. (See Fig. 88.) Then 85.6 grains minus 7.9416 grains equals 77.7 grains saturated. This moisture vapor is almost the same value as the number of grains of moisture vapor for air at 80°F. and .50 relative humidity, the difference being only .2. The chart, Fig. 88, shows the grains for 82.5°F. and 50% relative humidity and 77.9 grains. For simplicity, with such a small difference, the 77.9 grain line on the chart will be assumed as 77.7 here. This moisture vapor quantity is about 60°F. wet-bulb temperature and has a heat content of approximately 26.34 B.t.u. per pound of air.

The difference between 32.7 B.t.u., the heat content for the air at 82.5°F. and .505 relative humidity and the 26.34 B.t.u., the heat content for air at 60°F. and saturated, is $32.7-26.34=6.36$ B.t.u. per pound of air. Multiplying 6.36 by 24,000 pounds of air equals 152,640 B.t.u. per hour which is the heat transfer from the air to the water of the washer for maintaining 80°F. and 50 per cent relative humidity in the residence.

To determine the *actual* heat quantity required, the efficiency of the washer as a heat transfer unit, is considered. Most washers have an efficiency between 75 and 90 per cent. A washer with two banks of coils should have an efficiency of nearly 90 per cent. Thus the *actual* quantity can easily be determined from rated efficiencies of washers. This process, being simple, does not require a formula.

Solution 1E. The air washer will require about 6 gallons* of water per minute per 1000 cubic feet of air. The 6 gallons multiplied by 8.34 pounds (weight of one gallon of water) equals 50.04 pounds of water per minute per 1000 cubic feet of air passing through the air washer. The air at 80°F. and 50 per cent relative humidity has a volume of 13.86 cubic feet (this can be calculated from the chart or it can be found in Fig. 88) of air per pound. The 400 pounds of air per minute is multiplied by 13.86 which equals 5,540 cubic feet of air per minute. Then 5,540 is divided by 1000 which equals 5.54 thousands of cubic feet of air. This 5.54 multiplied by 50.04 equals 277.2216 pounds of water per minute. Multiplying 277.2216 by 60 minutes in one hour equals 16,633.296 pounds of water per hour for the air washer.

The 16,633.296 pounds of water per hour will remove 152,640 B.t.u. per hour from the air passing through the air washer.

The temperature difference of the water entering and leaving the air washer is determined by dividing 152,640 B.t.u. by 16,633.296 (call it 16,633.3) which equals 9.16°F. the temperature difference of the water in the air washer.

The above answer can also be obtained by substituting in the following formula.

$$T^L=\frac{H^T}{W} \tag{51}$$

*These capacities are given in manufacturers' data.

where H^T = total heat units necessary for humidification in the air washer.

T^L = the temperature loss of the water or the temperature drop of the inlet and leaving water of the air washer.

W = the pounds of water for the air washer per hour.

Substituting

$$T^L = \frac{152{,}640}{16{,}633.3}$$

$$T^L = 9.16$$

The water leaving the air washer is at the same temperature as the air, or 60°F. (air washer leaving water is always at the temperature of air in any application). To determine the entering temperature of the water, subtract 9.16°F. from 60°F. which equals 50.84°F. This is the temperature of entering water required to maintain the required conditions of 80°F. and 50 per cent relative humidity.

This same answer can be obtained from the following formula.

$$T^E = T^L - T^D \tag{52}$$

where T^E = the temperature of the water that is entering the air washer.

T^L = the temperature of the water that is leaving the air washer.

T^D = the temperature drop or difference of water temperature entering and leaving the air washer.

Substituting

$$T^E = 60° - 9.16°$$

$$T^E = 50.84°\text{F.}$$

To determine the *actual* temperature difference of the water entering and leaving the air washer, for cooling and dehumidifying, the efficiency of the air washer, as a heat exchange apparatus, must be considered. As a typical example, we assume the air washer efficiency is 75 per cent. Manufacturers' data or catalogues list actual efficiencies of their products.

The 152,640 is divided by 75 per cent which equals 203,520 B.t.u. per hour that is required for cooling and dehumidification. Then 203,520 divided by 16,633.3 equals 12.23°F. the *actual* temperature drop of the water passing through the air washer.

As previously explained, the water leaving the air washer is at the same temperature as the air leaving the air washer, or, in this case 60°F. To determine the *actual* entering temperature of the water for the air washer, subtract 12.23°F. from 60°F. which equals 47.77°F.

Example 2. A commercial building will require 120,000 pounds of air per hour for heating and ventilating of the interior. The entire air supply will be from the exterior. This means no recirculation will be carried on. The temperature of exterior air passing to the heating equipment can be assumed as 25°F. and 80 per cent relative humidity. The interior of the building is to be maintained at 65°F. and 45 per cent relative humidity. An air washer is to be used to clean and humidify the air. The occupants and various servicing and storage facilities of the building will provide 1,800,000 grains of moisture vapor per hour to the air of the interior.

2A. What is the quantity of moisture vapor that is necessary for proper humidification of the interior of the building?

2B. What is the total heat transfer for the moisture vapor application for humidification and heating of the air for the interior of the building?

2C. What is the weight of water required for the air washer for humidification and cleaning of the air?

2D. What is the temperature of the water that is required for the air washer to humidify and clean the air? Assume the air washer operates at 75 per cent thermal efficiency.

Note: The heat transfer required for increasing the air temperature and for vaporization of water for moisture vapor application is provided by preheating radiation and by the water of the washer as a heat transfer medium. The heat transfer required for warming the air to its final temperature is provided by reheating coils.

Solution 2A. By using the Psychrometric Chart found in the back of the book (Fig. 89 is an abbreviated chart showing points and lines used in this example), determine the grains of moisture vapor per pound of air for a temperature of 25°F. and a relative humidity of 80 per cent. This is shown by line 1 in Fig. 89 and is 15.26 grains of moisture vapor per pound. Line 2, in Fig. 89, shows how to find 41.67 as the number of grains of moisture for 65°F. air at 45 per cent relative humidity.

The difference between the moisture quantities 41.67 and 15.26 equals 26.41 grains of moisture vapor. The 26.41 multiplied by 120,000 pounds of air per hour equals 3,169,200 grains of moisture vapor required. The occupants and various storage and servicing conditions provide 1,800,000 grains per hour. To determine the amount of moisture vapor to be added to the air in the air washer and to have the total amount required per hour (3,169,200 grains), it is necessary to subtract 1,800,000 grains from 3,169,200 grains which equals 1,369,200 grains per hour. This is the amount to be mixed with the air for proper humidification. The 1,369,200 divided by 7,000 grains per pound equals 195.6 pounds of moisture vapor to be provided from the air washer for the required humidification of the interior.

The following formulas can be used to obtain the same results as just calculated.

$$G^A = G^T - G^1 \tag{53}$$

$$P^M = \frac{G^T}{7{,}000} \tag{54}$$

where G^A = grains of moisture vapor added to the air for humidification.

G^T = total grains of moisture vapor required for humidification of the air.

G^1 = total grains of moisture vapor from the interior spaces that is mixed with the air.

P^M = pounds of moisture vapor for the humidification of the air for the interior.

7,000 = grains of moisture in one pound.

The reader should be able to substitute in the formulas, as was done in Example 1.

Solution 2B. To determine the total heat (B.t.u.) required for the humidification and heating of the air for the interior, we again make use of the Psychrometric Chart. For the 25°F. and 80 per cent relative humidity air, the total heat is 7.6 B.t.u., as shown by line 3 in Fig. 89. In like manner the total heat of 65°F. and 45 per cent relative humidity interior air contains 21.97 B.t.u., as shown by line 4 in Fig. 89.

The difference between 21.97 B.t.u. and 7.6 B.t.u. equals 14.37 B.t.u. for the humidification of the air for the interior.

The following formula can be used to obtain the same results as were obtained in the previous calculations.

$$H^P = \text{B.t.u.}G^F - \text{B.t.u.}G^O \qquad (55)$$

where H^P = B.t.u. or heat units for providing the quantity of moisture vapor for application to the air by humidification.

G^F = grains of moisture vapor for the humidification of the air for the interior per pound of air.

G^O = grains of moisture vapor of the air entering the heating system per pound of air.

B.t.u. = heat units of the expressed grains of moisture vapor.

The 14.37 B.t.u. multiplied by 120,000 pounds of air per hour equals 1,724,400 B.t.u. per hour for the humidification and heating of the air in the commercial building.

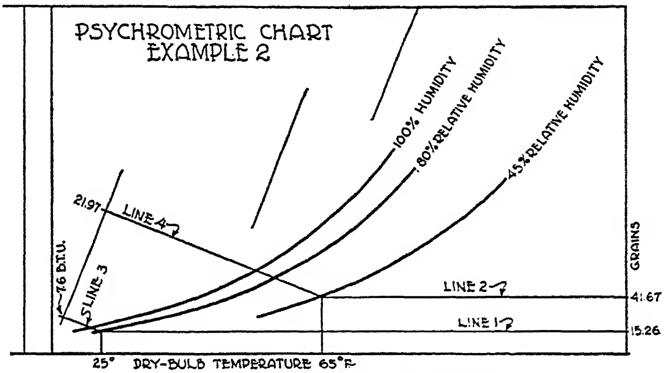


Fig. 89. Abbreviated Psychrometric Chart

This can also be calculated by using the following formula.

$$H^T = W^A \times H^{PA} \qquad (56)$$

where H^T = the total heat units necessary for moisture vapor application for humidification of the air.

W^A = the pounds of air per hour passing to and through the air washer.

H^{PA} = the heat units necessary for the moisture vapor per pound of air entering the interior spaces from the air washer.

Solution 2C. The water quantity for cleaning and humidification of the air in the air washer is usually from 4 to 8 gallons per minute per 1000 cubic feet of air per minute.

The 120,000 pounds of air per hour is divided by 60 minutes in one hour and multiplied by 12.26 cubic feet of air per minute.

Note: The 12.26 cubic feet of air for 25°F. and 80 per cent relative humidity is obtained by interpolation as between 24° and 26°F. dry and saturated condition of the air and multiplying the difference between 25°F. dry and 25°F. saturated conditions by 80 per cent. Then add this amount to the cubic feet volume of the air at 25°F. dry air. This equals the cubic feet of air at 25°F. and 80 per cent relative humidity.

The calculations are as follows:

$$12.24 = \text{dry air at } 26°\text{F.}$$
$$\underline{12.19} = \text{dry air at } 24°\text{F.}$$
$$.05 = \text{difference}$$
$$.05 \div 2 = .025$$
$$12.19 + .025 = 12.215 = \text{dry air at } 25°\text{F.}$$

Also

$$12.30 = \text{saturated air at } 26°\text{F.}$$
$$\underline{12.24} = \text{saturated air at } 24°\text{F.}$$
$$.06 = \text{difference}$$
$$.06 \div 2 = .03$$
$$12.24 + .03 = 12.27 = \text{saturated air at } 25°\text{F.}$$

Then $12.27 - 12.215 = .055$ difference

and $.055 \times .80 = 12.259$ or 12.26

This process of interpolation is quite simple after the reader has performed it a few times.

The 24,520 cubic feet of air per minute divided by 1000 equals 24.52 thousands of cubic feet of air per minute. Allowing 4 gallons per minute per 1000 cubic feet of air and multiplying 24.52 by 4 equals 98.08 gallons per minute for the air washer. Then 98.08 multiplied by 8.34 pounds of water per gallon equals 817.9872 pounds of water per minute or 817.9872 multiplied by 60 minutes in one hour equals 49,079.232 pounds of water per hour that is required for the air washer.

The following formula can also be used to obtain the weight of water.

$$W = \frac{H^T}{T^D} \tag{57}$$

where W = pounds of water for the air washer per hour.

H^T = total heat units necessary for humidification in the air washer.

T^D = the temperature loss of the water or the temperature drop of the inlet and leaving water of the air washer.

Solution 2D. The moisture vapor quantity that is required for the humidification of the air for maintaining 45 per cent humidity with 65°F. temperature is 195.6 pounds per hour. (See Solution 2A.) The heat transfer for the moisture vapor application to the air requires 1,055.2 B.t.u. per pound of water that is required to be evaporated.

Note: The 1,055.2 is calculated from a formula shown in the section on "Air Conditioning Principles" and is

$$LH = 1091 - .56t$$

where LH = latent heat

t = temperature

Substituting

$$LH = 1091 - .56 \times 65$$
$$LH = 1055.2$$

The 1,055.2 B.t.u. per pound multiplied by 195.6 equals 206,397.12 B.t.u. per hour required for humidification of the air at 65°F. and 45 per cent relative humidity.

The air washer as a heat transfer apparatus has an efficiency, we assume, of 75 per cent. To determine the heat transfer quantity that is required for the water of the air washer to provide the humidification, the 206,397.12 is divided by 75 per cent which equals 275,196.2 B.t.u. per hour. This is the B.t.u. required for heating the water to provide moisture vapor application to the air in the commercial building.

To determine the temperature of the water used for cleaning and humidifying the air, the 275,196.2 B.t.u. is divided by 49,079.232 pounds of water per hour, which equals 5.67°F. drop or difference of the temperature of the water entering and leaving the air washer. The water and the air leaving the air washer are always at the same, or approximately the same, temperature. To determine the temperature of the water entering the air washer, add the 5.67°F. to 65°F., leaving water temperature, which equals 70.67°F.

The above calculations can be carried on by the following formula.

$$T^E = \frac{H^T}{W} + T^F \tag{58}$$

where T^E = the temperature of water for the air washer in use for humidification of air.

H^T = total heat units necessary for humidification in the air washer.

W = the pounds of water for the air washer per hour.

T^F = the leaving or final temperature of the water following contact with the air in the air washer.

Note: In the various calculations of heat and moisture vapor application, the moisture vapor from any source, as occupants, equipment, etc., should be considered and deducted from the amount that would be the total moisture vapor required from the air washer. This will lower the quantity of heat exchange and the moisture vapor that is required from the air washer. As the moisture vapor in the air of the interior spaces will be readily mixed with the air passing to the interior from the heating system, the above sentence is true. The water temperature of the air washer should always be above the wet-bulb temperature of the air. Otherwise heat transfer or exchange cannot be available for vaporization of the water in contact with the air for moisture application. The heat for vaporization of water is from the water flowing through the spray nozzles of the air washer.

PRACTICE PROBLEMS

1. A restaurant to be air conditioned requires 50,000 pounds of air per hour for cooling and dehumidification. Thirty-five per cent of this air is exhausted to the outside from the kitchen and sixty-five per cent of the air for the interior is recirculated to the air-conditioning equipment. The interior of the restaurant is maintained at 80°F. with 50% relative humidity. The outside air is 95°F. with 60% relative humidity. The recirculated air is 80°F. and 50% relative humidity. The restaurant has a seating capacity for 80 people and it can be assumed that each person will eliminate 800 grains of moisture vapor per hour. There are 20 employes, such as waitresses or waiters, all of which will be very active and can be assumed to eliminate 1,200 grains of moisture per hour per person. The cooking and service equipment will provide 80,000 grains of moisture vapor per hour to mix with the air of the interior.

An air washer is to be used for cooling and dehumidification of the air in the interior of the restaurant.

1A. What is the weight of the air from the outside and the inside spaces entering the air-conditioning system?

1B. What is the temperature of the air mixture from the exterior and the interior spaces that is entering the air-conditioning system?

1C. What is the heat transfer for the cooling of the air? What is the heat transfer for the dehumidification of air for the interior of the restaurant?

1D. What is the quantity of moisture vapor that is removed by the heat transfer from the air and the water of the air washers for the dehumidification of the air in the interior of the restaurant?

1E. What is the weight of moisture vapor per hour that is removed from the air passing through the air washer?

1F. What is the weight of the water used in the air washer for the cleaning, cooling, and dehumidification of the air for the interior of the restaurant?

1G. What is the entering temperature of the water for use in cooling and dehumidification of the air for the interior?

1H. Describe the division of heat transfer as for cooling and dehumidification requirements of the air for the interior of the restaurant.

2. A six-room residence requires 14,020 pounds of air per hour for heating and ventilating the interior. The residence has six occupants and is to be maintained at 70°F. and 45% relative humidity. The occupants and the various servicing conditions will provide 10,000 grains of moisture vapor per hour to mix with the air passing from the heating and ventilating system to the interior. The air for heating and ventilating will have 25% of the total air supply required from the exterior and 75% recirculated from the interior. The outside air temperature is 30°F. and 75% relative humidity. The recirculated air is 60°F. and 40% relative humidity. An air washer is used for humidification of the air for the interior of the residence.

1A. What is the weight of the air from the exterior and interior spaces per hour for heating and ventilating the interior of the residence?

2B. What is the temperature of the mixture of air from the outside and inside spaces passing through the heating and ventilating system?

2C. What is the quantity of moisture vapor entering the heating and ventilating system per pound of air?

2D. What is the quantity of moisture vapor for humidification per pound of air, using the air washer as a humidifier?

2E. What is the heat transfer per pound of air and moisture vapor mixture for the humidification of the air of the residence?

2F. What is the temperature of the water of the air washer for the humidification of the air for the interior spaces of the residence?

Fans. There are numerous makes and kinds of fans used in mechanical furnaces any of which can be selected from manufacturers' catalogues once the requirements are known.

As previously explained, the air in a room must be changed. The amount or number of changes vary. But the total volume per hour of needed air can be calculated from the cubic contents of a given residence, for example, and multiplied by the number of changes required to find the total volume. Suppose for instance that an enclosure of 7,200 cubic feet requires 2 air changes per hour. This means a change of air every 30 minutes. We then divide the cubic contents of the enclosure by 30 minutes, which gives 240 cubic feet per minute. The fan selected must deliver not less than 30 per cent more air than that figured, because of friction losses and restrictions within the equipment, duct system, filter, etc. (See Technical Code,

Chapter VI.) Manufacturers' catalogues can be consulted to select proper fan rating.

For air-conditioning work the fan speed should be kept as low as possible. High-speed fans produce more noise than is desirable in air-conditioning systems.

Another way to select a fan, which is more accurate, is to choose a fan from the manufacturers' rating tables which is capable of delivering a volume of air, expressed in cubic feet per minute, against a frictional resistance, expressed in inches of water, computed by adding together the following items, values for which are given in the manufacturers' catalogues.

1. The frictional resistance of a warm air trunk or leader.
2. The frictional resistance of a return air trunk or duct.
3. The resistance to the flow of total volume of air through the furnace casing or hood, which is usually considered from 0.10 to 0.15 inches of water.
4. The frictional resistance through any other accessories, such as washers or filters.
5. A factor of safety of 10 per cent of the resistance calculated above.

Furnaces. Like all other equipment, furnaces are made in many shapes and sizes. However, their ratings are published by their manufacturers in tables such as Table 63 and their selection depends on requirements as determined by using the Technical Code.

Motors. Manufacturers generally specify the sizes of motors to be used with their equipment as shown in Table 63. Other apparatus needing motors generally are equipped with the required size.

Eliminators. Eliminators are generally an integral part of an air washer and need not be selected as a separate item.

Compressors. No positive summer air conditioning is possible without some form of refrigeration. Many systems employing ice, well water, etc., are in use but the most practical method is by artificial refrigeration.

A refrigerating machine is simply an apparatus that raises a given amount of heat to a higher temperature so that it can be carried away by some such substance as water or air. Applying this principle to air conditioning the refrigerating apparatus takes heat out of water, such as used in air washers, or takes heat out of air passing

*Table 63. Ratings Dailaire 100-200 Series for Coal

Outfit No.	Combustion Rate	B.t.u. at Bonnet	Safe B.t.u. Load	Gravity Rating in Pipe Area	C.f.m. Rating ¼" S.P.	Motor Size	Grate Area	Heating Surface	Ratio G.A. to H.S.†
124	7½	106,000	79,000	580	.....	..	324	6,797	21.0
124B1	7½	124,000	96,000	.....	1,000	¼	324	6,797	21.0
124B2	7½	124,000	96,000	.....	1,800	¼	324	6,797	21.0
224	7½	118,000	87,000	640	.....	..	324	10,052	31.0
224B1X	7½	132,000	106,000	.....	1,500	¼	324	10,052	31.0
228	7½	148,000	109,000	800	.....	..	396	11,116	28.0
228B2	7½	161,000	138,000	.....	1,800	¼	396	11,116	28.0
228B4	7½	161,000	142,000	.....	3,600	½	396	11,116	28.0
240	7½	181,000	157,000	1,156	.....	..	612	14,683	24.0
240B2	7½	248,000	210,000	.....	1,800	¼	612	14,683	24.0
240B4	7½	248,000	220,000	.....	3,600	½	612	14,683	24.0
2840B4	7½	304,000	260,000	.....	3,600	½	748	15,683	21.0
2840B4X	9	365,000	312,000	.....	6,000	1	748	15,683	21.0
3240B6X	9	430,000	366,000	.....	9,000	1½	884	16,179	18.2

*This is a sample rating table. Such tables can also be obtained for gas and oil and from all manufacturers. This particular table applies to a make, such as illustrated in Fig. 58.
†G.A. = grate area.
H.S. = heating surface.

Additional Data on 100 and 200 Coal Series

Note—To simplify this table, number of washer nozzles, number of filters and filter sizes are listed under the blower unit. These are furnished only under their respective outfits—see table below.

Outfit No.	124B1	124B2	224B1	228B2	228B4	240B2	240B4	2840B4	2840B4	3240B4	3240B6
Blower Size	No. ⅞	No. ⅞	No. 1	No. ⅞	No. ⅞	No. ⅞	No. ⅞	No. ⅞	No. 1	No. 1	No. 1
Std. Pulley Size	4	4	4	4	4	4½	4½	4½	4½	4½	4½
No. Spray Nozzles	2	2	2	4	4	4	4	6	6	...	...
No. Dry Filters	4	4	4	4	6	6	6	6	8	...	...
Size Dry Filters	16x20	16x20	16x20	20x20	16x25	16x25	16x25	16x25	16x25	...	...
Size Chimney	8x8	8x8	8x8	8x12	8x12	8x12	8x12	12x12	12x12	12x12	12x12

through cooling coils and transfers the heat to the above mentioned air or water. In large air-conditioning systems the heat is discharged to water and the refrigerating machine is called a water-cooled machine.

It is not within the scope of this book to give a full explanation of refrigeration principles other than as applied to air conditioning.

The *ton of refrigeration* is a term used in rating refrigerating machines and determining how much refrigeration a given system requires.

The total B.t.u. figured for a given residence, etc. (cooling load), divided by 12,000 gives the size or tonnage of refrigeration required (12,000 represents B.t.u. per ton). Thus if the cooling load for a residence is 24,000 B.t.u. then $\frac{24{,}000}{12{,}000} = 2$ tons of refrigeration required. One ton of refrigeration equals 12,000 B.t.u. per hour or 200 B.t.u. per minute.

Thus a refrigerating machine rated at 2 tons, or nearly so, would be required to handle a system where the cooling load is 24,000 B.t.u. per hour.

Table 64 shows a typical rating table for refrigerating machines.

TABLE 64. CARRIER REFRIGERATING MACHINES

Types 7F5, 7H5, 7F6, 7H6, 7F66 (Water or Evaporative Condensers)

These units, water or evaporative condenser cooled, and using either Methyl Chloride or Freon as refrigerant, represent machines usable for medium small refrigeration loads, including air-conditioning service. The type 7F66 series duplex machine consists of two compressors and motors on a single base. An economy results from using the single machine than is had from using two machines each of which is one-half the duplex capacity.

Data on Freon Machines

	7F5-50	7H5-75	7F6-100	7H6-150	7F66-100-100
Motor Hp.	5	7½	10	15	Two-10
Std. Speed R.p.m.	450	600	450	600	450
Nom. Capacity*					
1000 B.t.u./Hr.	68.0	90.6	136.0	181	272
Tons	5.66	7.56	11.64	15.05	22.64
Connections, Ins.					
Suction, W.C.	1⅜	1⅝	2⅛	2⅝	3⅛
Liquid, E.C.	1⅜	1⅝	2⅛	2⅝	3⅛
Water in W.C.	1 F.P.T.	1¼ F.P.T.	1¼ F.P.T.	1½ F.P.T.	2 F.P.T.
Water out W.C.	1 F.P.T.	1¼ F.P.T.	1¼ F.P.T.	1½ F.P.T.	2 F.P.T.
Dimensions, Ins.					
Length W.C.	60½	59¾	61¼	63¾	105
Length E.C.	50	50	50	50	95¾
Width W.C.	30⅛	30	37⅜	37⅜	38¼
Width E.C.	26¼	26¼	35½	35½	33 7/16
Height W.C.	41¾	44 7/16	41¾	44 7/16	46 5/16
Height E.C.	30¾	33½	30¾	33½	32¾
Net Weight W.C.	1175	1265	1840	1895	2650
Net Weight E.C.	1030	1120	1435	1535	

*Suction 40° F., Condensing 98° F.

Note 1. W. C.—water cooled.
E. C.—evaporation condenser.

Note 2. Evaporative cooler is an apparatus to take the place of water for cooling.

Note 3. Other tables and data on evaporative condensers can be obtained from manufacturers.

Calculation of Cooling Loads. In a previous section, the main items considered in cooling load calculations were given together with several tables and illustrations all of which formed a general discussion of cooling loads. In this section we are ready to apply cooling load principles to a definite example.

To calculate the cooling load, it is easier to consider the problem in two divisions, namely, the "Cooling Load" and "Dehumidifying Load." These two divisions can be broken up into the following items.

Cooling Load:

1. Transmission through walls, roofs, glass, doors, etc. is like calculating heat losses as explained for Formula (**1**), Vol. II, and to calculate transmission, Formula (**14**), Vol. II, is used. (See also Chapter V, Vol. II.) It should be remembered that an allowance must be made for solar radiation as explained under "Heat Gains."

2. Radiation of sun through windows. This is explained in a previous section.

3. Infiltration and ventilation.

4. Heat from occupants. This item may be almost negligible in sparsely occupied premises, but very important where groups of occupants are concerned.

5. Heat from electric lights and any cooling or other processing.

Dehumidifying Load:

1. The moisture brought in by infiltration of outside air or by ventilation. The infiltration may be neglected where ventilation of so many cubic feet of air per minute per occupant is supplied.

2. Moisture from the occupants of the premises.

3. Moisture from cooling or other processing being carried on within the conditioned space.

The application of these items to the estimating of cooling loads is less exact than the calculation of heat losses. This is due to the fact that some of the methods are simple estimates rather than exact mathematical calculations. However, the methods used are recommended rather than to have the heating and ventilating engineer troubled with cumbersome mathematics. The results obtained are accurate within reason. The reader must be thoroughly

familiar with the principles of air conditioning which are explained in Chapter IV.

To illustrate the practical application of cooling load calculations, a typical illustrative example is given in the following.

Example. Fig. 90 shows a sketch for a living room of a large residence. The residence outside walls are of 12-inch brick walls. The plaster is on 1-inch rigid insulation which is furred out from the wall. The space above the room is

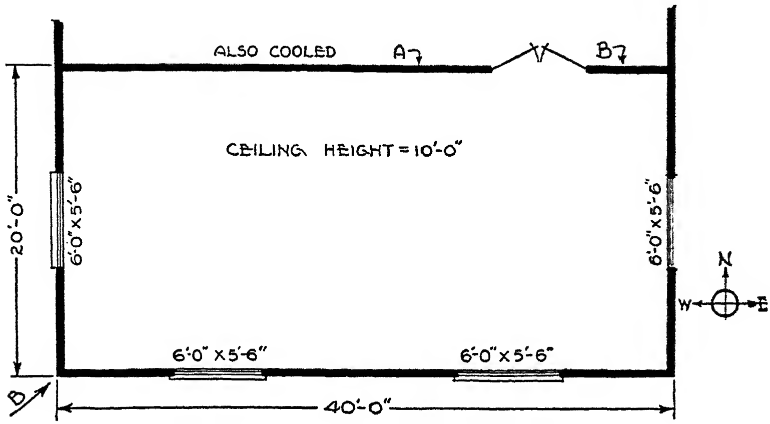


Fig. 90. Plan View of the Living Room of a Residence

not cooled and can be assumed as being the same as outside temperature. The space under the room is the basement which can be assumed as being 10°F. cooler than the outside temperature. The balance of the same floor is also cooled so the partition *A* and door *B* need not be considered. The ceiling construction is 2x10 joists with ½-inch rigid insulation used as lath. The floor is oak finish on yellow pine sub-floor. The floor over the basement is 2x10 joists and oak finish floor on yellow pine sub-flooring. There is no plaster for the basement ceiling. The windows are single glass. Assume total lighting is 500 watts.

Calculate the cooling load if the outside temperature is 95°F. dry bulb and 75°F. wet bulb and the desired inside temperature 80°F. dry bulb and 65°F. wet bulb. Assume 30 as greatest number of people at any one time.

Solution. The first steps in the solution are to calculate the transmission gains through walls, windows, ceiling, and floor. To calculate such gains, we must consider sun effect as shown in Table 28 and as previously explained. The sun does not shine on all three walls at one time so it stands to reason that sun effect need not be considered for all wall surfaces. The east wall can be eliminated, as far as sun effect is concerned, because the sun's rays are upon it a comparatively short time. The south and west walls receive the full effect of the sun's rays so these walls must be carefully considered. The south wall receives less intense rays from the sun than the west wall receives. However, the south wall has twice the area of the west wall and contains double the window area. Calculation would show that the south wall has a greater sun effect load than the west wall.

Therefore the sun effect will be considered only on the south wall in calculating the cooling load. The reader should understand that we are calculating the *maximum* heat gain that will occur at any one time. This is done so that the answer, in tons of refrigeration, will represent the greatest load that the cooling apparatus will be called on to carry. This enables the selection of apparatus having the proper capacity. When the sun reaches a point indicated by the arrow at B, in Fig. 90, its rays strike both the south and west walls. However, at this point the angle of the rays is such, on both walls, that no greater load is assumed than that for the south wall alone.

The walls are of brick. Therefore, from Table 28, we see that it is necessary to add 10° to the calculated temperature difference to include sun effect in the heat gain calculation. The temperature difference is 95°–80° = 15°F. This 15°F. temperature difference will be used in calculating heat gains through the east and west walls. For the south wall (including sun effect) 10° is added to the 15° making the temperature difference 25°F.

South Wall:

Total area $=40\times10 = 400$ square feet.
Glass area $=2(6\times5\frac{1}{2}) = 66$ square feet
Net wall area $=400-66=334$ square feet.
U value for wall $= .14$ (See Table 3, Vol. II).
U value for glass $=1.13$ (See Table 13, Vol. II).
Temperature difference $=25°$F.

Substituting in Formula (14), Vol. II

(Wall) $H_t=334\times.14\times25$
$H_t=1169$ B.t.u. per hour.
(Glass) $H_t=66\times1.13\times25$
$H_t=1{,}865$ B.t.u. per hour.

Note: Some engineers consider that for bare windows, on which the sun shines, the heat gain due to transmission is small as compared to the sun effect. As a result they disregard the transmission gain (such as 1,865 B.t.u. in this example) and consider only the sun effect or radiation gain (7,920 B.t.u. as calculated in the following under "Radiation through Glass") For windows not exposed to the sun the usual transmission gain is calculated. In this example both ***transmission*** and ***radiation*** gains are considered, which constitutes an ample factor of safety.

West Wall:

Total area $=20\times10=200$ square feet.
Glass area $=6\times5\frac{1}{2}= 33$ square feet.
Net wall area $=200-33=167$ square feet.
U value for wall $= .14$
U value for glass $=1.13$
Temperature difference $=15°$F.

Substituting in Formula (14), Vol. II

(Wall) $H_t=167\times.14\times15°$
$H_t=351$ B.t.u. per hour.
(Glass) $H_t=33\times1.13\times15°$
$H_t=559$ B.t.u. per hour.

East Wall:

This is exactly the same as the west wall.

(Wall) $H_t=351$ B.t.u. per hour.
(Glass) $H_t=559$ B.t.u. per hour.

Floor:

Total area $=40\times20=800$ square feet.
U value for floor $=.34$ (See Table 8, Vol.II).
Temperature difference $=95^\circ-85^\circ=10^\circ$F.

Substituting in Formula (14), Vol. II

$$H_t=800\times.34\times10$$
$$H_t=2{,}720 \text{ B.t.u. per hour.}$$

Ceiling:

Total area $=40\times20=800$ square feet.
U value of ceiling $=.16$ (See Table 8, Vol. II).
Temperature difference $=95^\circ-80^\circ=15^\circ$F.

Substituting in Formula (14), Vol. II

$$H_t=800\times.16\times15$$
$$H_t=1{,}920 \text{ B.t.u. per hour.}$$

Infiltration:

It can be assumed that 10 cubic feet per minute per person is supplied by the fan.

$$10\times30\times60=18{,}000 \text{ cubic feet per hour.}$$
$$\frac{18{,}000\times(95-80)}{55.2^*}=489 \text{ B.t.u. per. hour}$$

Radiation through Glass: We assume the sun shining only on the south wall. From Fig. 15 in Vol. II the south wall has an hourly (maximum is used) value of 120 B.t.u. per square foot. The south windows $=2(6\times5\frac{1}{2})=66$ square feet.

$66\times120=7{,}920$ B.t.u. per hour.

Occupants: There are 30 occupants who may be assumed as seated at rest. It is seen that for a temperature of 80°F. the sensible heat loss is 220 B.t.u. per hour per person.

Therefore $30\times220=6{,}600$ B.t.u.

Lighting $=500\times3.415=1.708$ B.t.u.

Dehumidifying Load: The next step is to calculate the dehumidifying load.

The outside air at 95°F. dry bulb and 75°F. wet bulb has a moisture content of 98 grains per pound (see Psychrometric Chart). The room air at 80° dry bulb and 65°F. wet bulb has a moisture content of 68 grains per pound. This gives a difference of 30 grains per pound to be removed. The volume of 1 pound of air at room conditions is 13.8 cubic feet. (See Psychrometric Chart.)

$$\frac{18{,}000}{13.8}\times\frac{30}{7{,}000}=1{,}304\times.0043=5.61 \text{ pounds per hour.}$$

The moisture given off by each person for the conditions would be 1,200 grains per hour.

$$30\times\frac{1{,}200}{7{,}000\dagger}=5.10 \text{ pounds per hour.}$$

Then the whole moisture load is $5.61+5.10=10.71$ pounds per hour. The latent heat is $1{,}092-(.56\times80)=1{,}047$ per pound. Thus the dehumidifying load (latent heat) is

$$10.71\times1{,}047=11{,}213 \text{ B.t.u.}$$

*A constant.
†Moisture per pound.

The total load or sum of the cooling and dehumidifying loads equals the load put upon the refrigerating apparatus. This can be summed up as follows.

Cooling Load:

South wall (Transmission)	1,169 B.t.u.
West wall (Transmission)	351
East wall (Transmission)	351
Floor (Transmission)	2,720
Ceiling (Transmission)	1,920
Infiltration	489
Radiation	7,920
Occupants	6,600
Glass (South) (Transmission)	1,865
Glass (East) (Transmission)	559
Glass (West) (Transmission)	559
Lighting	1,708
Total cooling load	26,211 B.t.u.

The total dehumidifying load = 11,213 B.t.u.
Then $26{,}211 + 11{,}213 = 37{,}424$ B.t.u. total load.
One ton of refrigeration = 12,000 B.t.u.
Then $37{,}424 \div 12{,}000 = 3.11$ tons of refrigeration cooling load.

With the above solution the reader should be able to understand easily the calculation of cooling loads. The various rooms in a residence, for instance, must be surveyed for cooling loads in the same manner that heat losses must be found for each room. Care should be exercised to see that the conditions surrounding each room are carefully studied. Entire structures may be cooled or only certain rooms. In either case the cooling loads can be calculated as indicated.

Special Note: The foregoing solution employs a method as near *standard* as our present-day knowledge of sun effect, etc., permits. The solution is simple and empirical. However, the results are accurate enough to merit its use.

In the field the various manufacturers and engineers practically all use different methods. This is because no standard, such as the Technical Code, has yet been devised.

The example presented in Chapter XIII gives a solution of a typical air-conditioning job calculated somewhat differently from the foregoing example and illustrates the thought given in the preceding paragraph.

Therefore the reader should not be alarmed at differences in methods. The two different methods presented herein are to show that such differences do exist. Fortunately most all methods give approximately the same results.

Cooling Methods. Air-conditioning furnaces may supply cooled air in typical ways as follows.

1. By air washer.
2. By cooling coils.
3. Silica Gel.
4. By natural air.

The cooling method whereby the air washer is used has already been explained.

Cooling by coils consists of forcing the air through coils, Fig. 91, in which cold water or a refrigerant is circulated. Fig. 92 shows

how such coils may be used in connection with a furnace system. The cooling coil follows the filter. The system shows a refrigeration

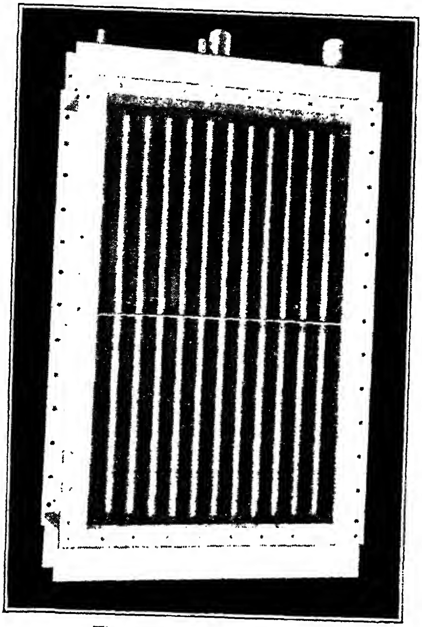


Fig. 91. Typical Cooling Coil
Courtesy of the Trane Company, La Crosse, Wisconsin

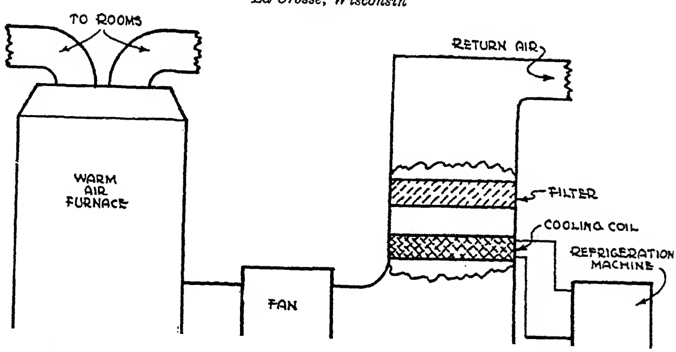


Fig. 92. Typical Air-Conditioning Furnace System Using a Coil for Cooling

machine to supply the coil. Other systems make use of refrigerated water or natural cool water which is below 50°. All such coils can be selected from the manufacturers' catalogues.

Selection of Cooling Coils. Cooling coils, as applied to air conditioning, using mechanical refrigerants as a cooling medium, encounter conditions so variable that to obtain maximum efficiency of a coil it must be designed for the condition under which it is to operate. Studies of heat transfer have proved that the velocity of a cooling medium through the coil greatly effects the rate of heat flow or capacity of cooling coils. If the velocity of the refrigerant is low, the heat transfer is low due to the stagnation of the refrigerant.

To simplify the selection of the proper coil, this section includes alignment charts, Figs. 93 and 94, that can be used to find the type of coil for any design after the total load and face area of the coil have been determined.

The rating Tables 65 and 66 give performance data for direct

†Table 65. Trane Rating Table

FACE VELOCITY 500 ft. PER MIN. 40° REFRIG. TEMP. Freon or Methyl Chloride 5° Superheat

Initial Wet Bulb	Rows of Tubes	INITIAL DRY-BULB TEMPERATURE														
		100			95			90			85			80		
		Final D.B.	Final W.B.	Tons*	Final D.B.	Final W.B.	Tons*	Final D.B.	Final W.B.	Tons*	Final D.B.	Final W.B.	Tons*	Final D.B.	Final W.B.	Tons*
80	1	90.0	76.1	0.71	86.3	76.1	0.71	82.6	76.1	0.71						
	2	81.4	72.1	1.38	78.8	72.1	1.38	76.3	72.1	1.38						
	3	74.9	68.6	1.93	73.2	68.6	1.93	71.2	68.6	1.93						
	4	69.6	65.3	2.40	68.3	65.3	2.40	67.0	65.3	2.40						
	5	65.0	62.2	2.81	64.0	62.2	2.81	63.2	62.2	2.81						
	6	61.1	59.4	3.17	60.5	59.4	3.17	59.8	59.4	3.17						
	7	57.9	56.8	3.47	57.5	56.8	3.47	56.9	56.8	3.47						
	8	55.2	54.6	3.72	55.0	54.6	3.72	54.6	54.6	3.72						
75	1	88.9	71.3	0.61	85.4	71.3	0.61	81.7	71.3	0.61	78.1	71.3	0.61			
	2	79.7	67.7	1.14	77.2	67.7	1.14	74.5	67.7	1.14	72.1	67.7	1.14			
	3	72.8	64.4	1.61	71.0	64.4	1.61	69.3	64.4	1.61	67.5	64.4	1.61			
	4	67.3	61.4	2.00	66.0	61.4	2.00	64.8	61.4	2.00	63.4	61.4	2.00			
	5	62.7	58.7	2.34	61.8	58.7	2.34	60.8	58.7	2.34	59.8	58.7	2.34			
	6	58.7	56.2	2.63	58.2	56.2	2.63	57.5	56.2	2.63	56.7	56.2	2.63			
	7	55.8	54.1	2.87	55.4	54.1	2.87	55.0	54.1	2.87	54.3	54.1	2.87			
	8	53.2	52.1	3.08	53.0	52.1	3.08	52.5	52.1	3.08	52.1	52.1	3.08			
70	1	*88.0*	*66.3*	*0.54*	84.0	66.5	0.51	80.4	66.5	0.51	76.7	66.5	0.51	73.3	66.5	0.51
	2	78.5	63.1	0.95	76.0	63.4	0.94	73.3	63.4	0.94	70.6	63.4	0.94	68.2	63.4	0.94
	3	71.1	60.2	1.35	69.3	60.5	1.32	67.5	60.5	1.32	65.5	60.5	1.32	63.7	60.5	1.32
	4	65.0	57.5	1.66	64.0	57.9	1.63	62.7	57.9	1.63	61.3	57.9	1.63	60.0	57.9	1.63
	5	60.1	55.0	1.94	59.7	55.5	1.91	58.7	55.5	1.91	57.7	55.5	1.91	56.8	55.5	1.91
	6	56.6	53.1	2.17	56.2	53.4	2.14	55.4	53.4	2.14	54.8	53.4	2.14	54.0	53.4	2.14
	7	53.8	51.3	2.36	53.5	51.5	2.34	53.0	51.5	2.34	52.4	51.5	2.34	51.8	51.5	2.34
	8	51.4	49.7	2.53	51.2	49.9	2.51	51.9	49.9	2.51	50.5	49.9	2.51	50.0	49.9	2.51
65	1	*88.0*	*60.9*	*0.54*	*84.0*	*61.2*	*0.50*	*80.0*	*61.6*	*0.45*	76.1	61.9	0.41	72.7	62.0	0.40
	2	*78.5*	*57.4*	*0.97*	*75.3*	*58.1*	*0.89*	*72.1*	*58.7*	*0.81*	69.0	59.1	0.76	66.5	59.1	0.76
	3	*70.8*	*54.3*	*1.31*	*68.2*	*55.3*	*1.21*	65.5	56.2	1.11	63.5	56.6	1.06	61.9	56.7	1.05
	4	*64.5*	*51.7*	*1.60*	*62.5*	*53.0*	*1.46*	60.5	54.0	1.35	59.3	54.4	1.31	58.0	54.4	1.31
	5	*59.6*	*49.6*	*1.82*	*58.0*	*51.0*	*1.67*	56.7	52.0	1.56	55.8	52.4	1.52	54.8	52.4	1.52
	6	*55.7*	*47.8*	*1.99*	54.2	49.4	1.83	53.8	50.4	1.74	52.8	50.6	1.71	52.3	50.7	1.70
	7	*52.6*	*46.4*	*2.13*	51.6	48.1	1.97	51.1	48.8	1.89	50.6	49.1	1.86	50.2	49.2	1.86
	8	*50.0*	*45.1*	*2.25*	49.4	46.9	2.08	49.1	47.5	2.02	48.9	47.8	1.99	48.6	47.8	1.99

*Tons per square foot of face area.
†Courtesy of the Trane Company, LaCrosse, Wisconsin.

†Table 66. Trane Rating Table

FACE VELOCITY **500 ft.** PER MIN. **45°** REFRIG. TEMP. Freon or Methyl Chloride 5° Superheat

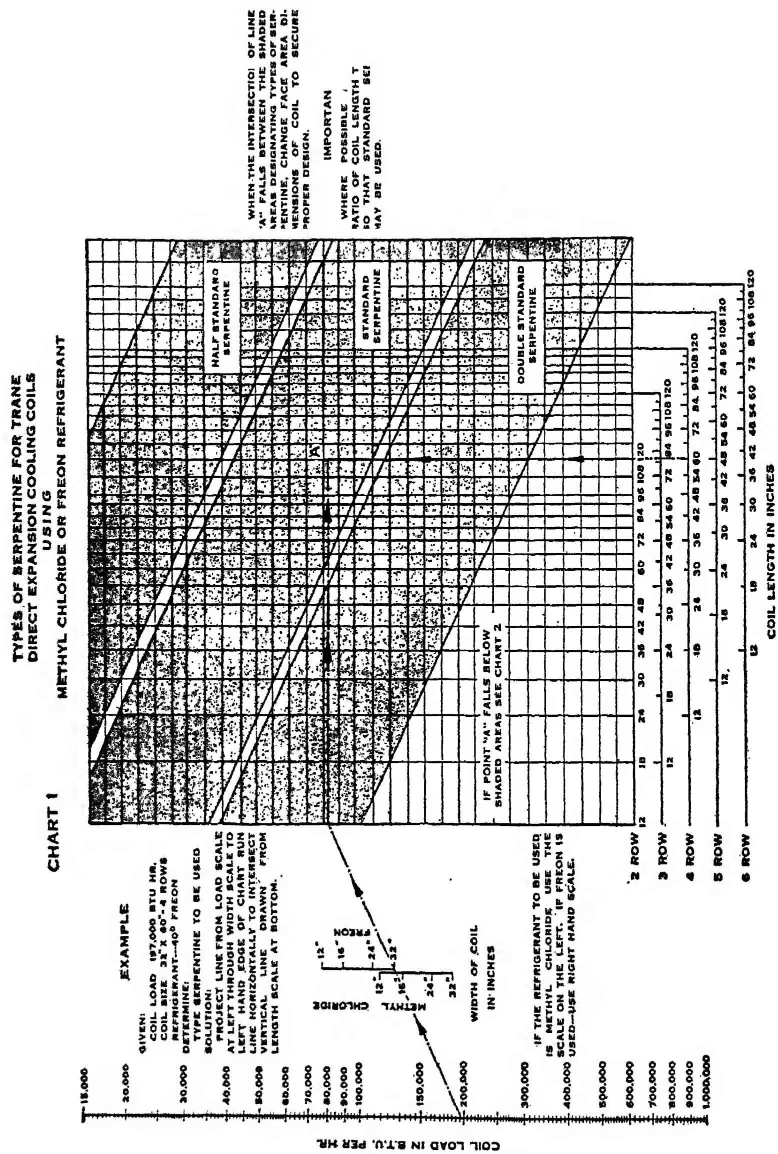
Initial Wet Bulb	Rows of Tubes	INITIAL DRY-BULB TEMPERATURE														
		100			95			90			85			80		
		Final D.B.	Final W.B.	Tons*	Final D.B.	Final W.B.	Tons*	Final D.B.	Final W.B.	Tons*	Final D.B.	Final W.B.	Tons*	Final D.B.	Final W.B.	Tons*
80	1	90.3	76.5	0.65	86.7	76.5	0.65	83.2	76.5	0.65						
	2	82.6	73.1	1.22	80.0	73.1	1.22	77.5	73.1	1.22						
	3	76.5	70.0	1.71	74.5	70.0	1.71	73.0	70.0	1.71						
	4	71.5	67.1	2.14	70.0	67.1	2.14	68.8	67.1	2.14						
	5	67.3	64.4	2.51	66.3	64.4	2.51	65.4	64.4	2.51						
	6	64.0	62.0	2.83	63.3	62.0	2.83	62.7	62.0	2.83						
	7	61.2	59.9	3.10	60.6	59.9	3.10	60.1	59.9	3.10						
	8	58.7	57.9	3.35	58.3	57.9	3.35	58.0	57.9	3.35						
75	1	89.0	71.7	0.55	85.4	71.7	0.55	81.9	71.7	0.55	78.3	71.7	0.55			
	2	80.7	68.7	1.00	78.3	68.7	1.00	75.5	68.7	1.00	73.1	68.7	1.00			
	3	73.5	65.8	1.41	72.4	65.8	1.41	70.5	65.8	1.41	68.7	65.8	1.41			
	4	69.4	63.3	1.76	67.9	63.3	1.76	66.6	63.3	1.76	65.1	63.3	1.76			
	5	65.0	61.0	2.06	64.2	61.0	2.06	63.2	61.0	2.06	62.3	61.0	2.06			
	6	61.9	59.0	2.31	61.1	59.0	2.31	60.4	59.0	2.31	59.8	59.0	2.31			
	7	59.0	57.1	2.53	58.5	57.1	2.53	58.0	57.1	2.53	57.5	57.1	2.53			
	8	56.7	55.4	2.72	56.4	55.4	2.72	56.0	55.4	2.72	55.6	55.4	2.72			
70	1	*89.0*	*66.6*	*0.50*	*85.0*	*66.9*	*0.45*	81.0	67.1	0.43	77.5	67.1	0.43	73.9	67.1	0.43
	2	*80.3*	*63.8*	*0.89*	77.3	64.3	0.82	74.3	64.4	0.80	71.8	64.4	0.80	69.2	64.4	0.80
	3	73.5	61.4	1.20	71.0	61.9	1.14	69.0	62.0	1.12	67.2	62.0	1.12	65.1	62.0	1.12
	4	67.5	59.2	1.47	66.0	59.7	1.41	64.9	59.9	1.39	63.5	59.9	1.39	62.2	59.9	1.39
	5	63.4	57.3	1.70	62.3	57.8	1.64	61.4	57.9	1.63	60.2	57.9	1.63	59.5	57.9	1.63
	6	59.8	55.7	1.88	58.9	56.0	1.85	58.5	56.2	1.83	57.7	56.2	1.83	57.1	56.2	1.83
	7	57.1	54.2	2.05	56.6	54.5	2.02	56.2	54.6	2.01	55.7	54.6	2.01	55.2	54.6	2.01
	8	55.0	52.9	2.19	54.5	53.2	2.16	54.3	53.3	2.15	54.0	53.3	2.15	53.6	53.3	2.15
65	1	*89.0*	*61.3*	*0.50*	*85.0*	*61.6*	*0.45*	*81.0*	*62.0*	*0.41*	*77.0*	*62.3*	*0.36*	73.1	62.5	0.33
	2	*80.3*	*58.1*	*0.89*	*77.1*	*58.7*	*0.81*	*73.8*	*59.4*	*0.73*	71.0	60.1	0.64	68.0	60.3	0.61
	3	*73.2*	*55.3*	*1.20*	*70.7*	*56.3*	*1.09*	*68.1*	*57.2*	*0.99*	65.5	58.1	0.88	63.7	58.3	0.86
	4	*67.5*	*53.0*	*1.46*	*65.5*	*54.2*	*1.33*	*63.4*	*55.4*	*1.20*	61.8	56.4	1.08	60.1	56.5	1.07
	5	*63.0*	*51.1*	*1.67*	*61.4*	*52.5*	*1.51*	*59.7*	*53.9*	*1.36*	58.7	54.8	1.26	57.6	54.9	1.25
	6	*59.4*	*49.5*	*1.83*	*58.1*	*51.1*	*1.66*	56.7	52.6	1.50	56.0	53.4	1.41	55.2	53.6	1.39
	7	*56.5*	*48.2*	*1.96*	*55.5*	*50.0*	*1.78*	54.5	51.6	1.61	54.0	52.2	1.54	53.7	52.4	1.52
	8	*54.2*	*47.1*	*2.06*	*53.4*	*49.0*	*1.87*	52.7	50.6	1.72	52.3	51.2	1.65	52.1	51.3	1.64

*Tons per square foot of face area.
†Courtesy of the Trane Company, LaCrosse, Wisconsin.

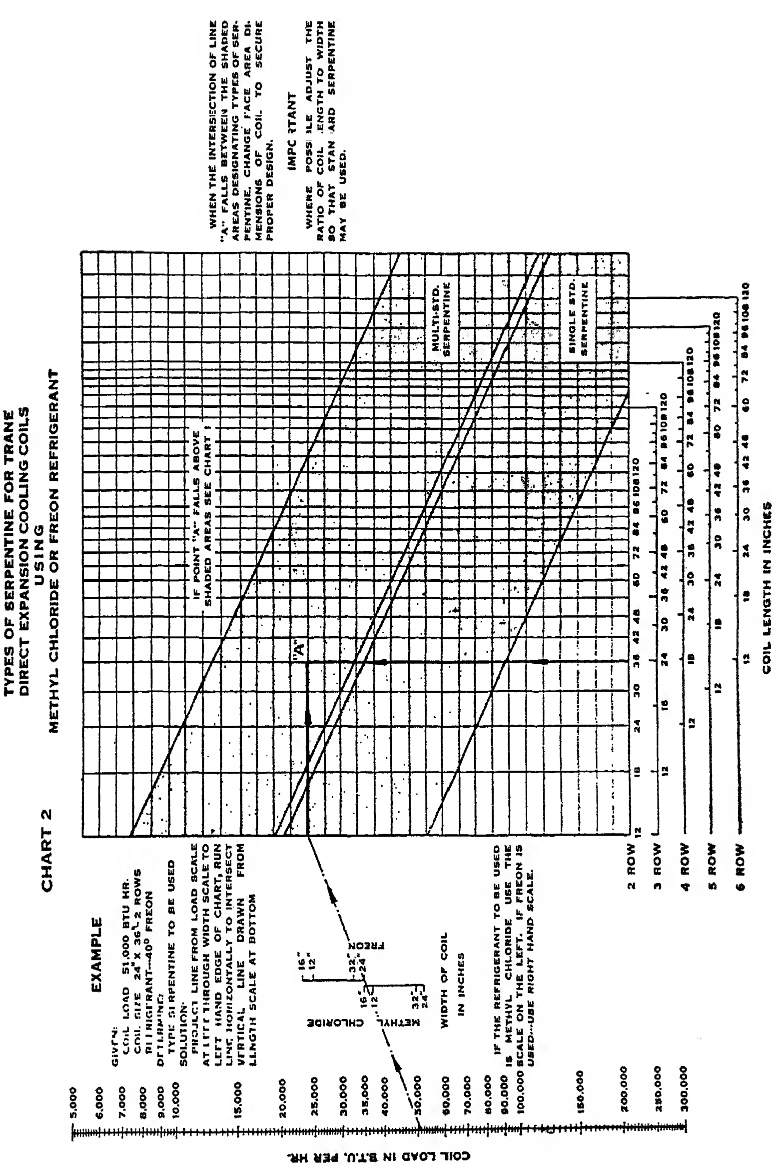
expansion coils with either Freon or Methyl Chloride. These tables are based upon the refrigerant vapor being superheated five degrees when leaving the coil.

Dry Surface. There can be no dehumidification in any cooling coil until the fins have been cooled to a point below the dew point temperature of the entering air. Therefore, it is quite possible for an initial portion of a coil to be dry; that is, no dehumidification will take place until the air has traveled some distance through the coil.

The rating tables show the dry portions of the coil. Figures in italics mean that that portion of the coil is dry; no dehumidification of the air is taking place. Thus, in Table 65 for a 40-degree refrigerant and 500 feet velocity, for an initial condition of 90 degrees dry



Figs. 93 and 94.
Courtesy of the Trane Company,



Alignment Charts
La Crosse, Wisconsin

bulb and 65 degrees wet bulb, the first two rows of coils will be dry; for initial condition of 95 degrees dry bulb and 65 degrees wet bulb, the first 5 rows will be dry; and for an initial condition of 100 degrees dry bulb and 65 degrees wet bulb, 8 rows will be dry.

A series of dots above the performance figures means that less than one row of the coil is dry. Thus, returning to the table for 40 degrees refrigerant and 500 feet velocity, for an initial condition of 95 degrees dry bulb and 70 degrees wet bulb, a series of dots is found above the figures giving the performance of a one-row coil. This means that a fraction of the first row is dry.

Use of the Tables. In selecting coils the final wet-bulb temperature needed is usually known. Use the tables for the desired refrigerant temperature and air velocity. Knowing the initial dry- and wet-bulb temperatures, find the figures representing the desired final dry- and wet-bulb temperatures. The number of rows of tubes needed to give this final condition is given in the left-hand column.

Example. Initial condition, 85 degree dry bulb and 70 degree wet bulb.

Final condition needed, 57.5 degrees dry bulb and 56.3 degrees wet bulb.

Air quantity, 10,000 C.F.M.

Face velocity 500 ft. per minute.

Turning to Table 66 for 45 degrees refrigerant and 500 face velocity, 6 rows of tubes are needed for a final dry-bulb temperature of 57.7 degrees and a wet-bulb temperature of 56.2 degrees, the values closest to the desired final condition.

The capacity of the 6-row coil is 1.83 tons per square foot of face area.

$$\text{Total face area required} = \frac{10{,}000}{500}$$
$$= 20 \text{ sq. ft.}$$

Total refrigerating capacity of coil $= 20 \times 1.83 = 36.6$ tons

Selecting Refrigerant Temperatures. The higher the refrigerant temperature used the lower will be the operating cost of the compressor. In addition, either a smaller compressor or a slower speed compressor may be used when a higher refrigerant temperature is used. However, the higher the refrigerant temperature selected, the

larger the number of rows of tubes which must be used in the cooling coil. It is always a question of balancing the increased cost of the larger coil against the decreased operating cost of the compressor.

It is interesting to select coils for various refrigerant temperatures for the example in the preceding section. The number of rows of tubes needed for the various refrigerant temperatures is given in Table 67.

Table 67. Tubes Needed for Various Refrigerant Temperatures

Refrig. Temp.	Rows of Tubes Needed	Final Dry Bulb	Final Wet Bulb	Tons Per Sq. Ft. of Face Area	Total Refrig. Capacity Tons
35	4	59.0	55.9	1.86	37.2
40	5	57.7	55.5	1.91	38.2
45	6	57.7	56.2	1.83	36.6
50	8	57.5	56.7	1.77	35.4

Interpolation. When ratings of coils are needed for conditions which are not given in the tables, interpolation between the tabulated values will give results which are sufficiently accurate for most problems.

The foregoing examples give a general description of selecting coils where a refrigerant is used as the cooling medium.

Some cooling coils have cold water as the cooling medium. The process of selection is somewhat the same.

The reader is advised to procure manufacturers' catalogues before attempting to select coils of either type due to the fact that improvements and other changes are constantly being made.

No more specific selection methods are given here because each manufacturer employs different methods which can best be secured from catalogues and instruction manuals.

Silica Gel Method. Fig. 95 shows a Silica Gel System where the incoming air is dehumidified and then cooled by coils. The silica gel absorbs moisture from the air leaving it dry and only needing slight cooling which can be secured from cold water or refrigerant coils.

Natural Cooling. By using natural air, some degree of cooling can be secured in several ways. One way to secure natural cooling is to employ a system such as shown in Fig. 92 without the cooling coil. During the summer, air is taken from basements or directly from the outside (no recirculation) and forced through a residence. This circulation of air gives a cooling sensation somewhat as does an elec-

tric fan. At night the cooler air is used in the same manner and the house can be cooled to within 80 per cent of the difference between the warm air inside and the cooler night air outside.

Another method of using cool night air is to put a fan in the attic of a residence so that it draws the cool air in through windows and

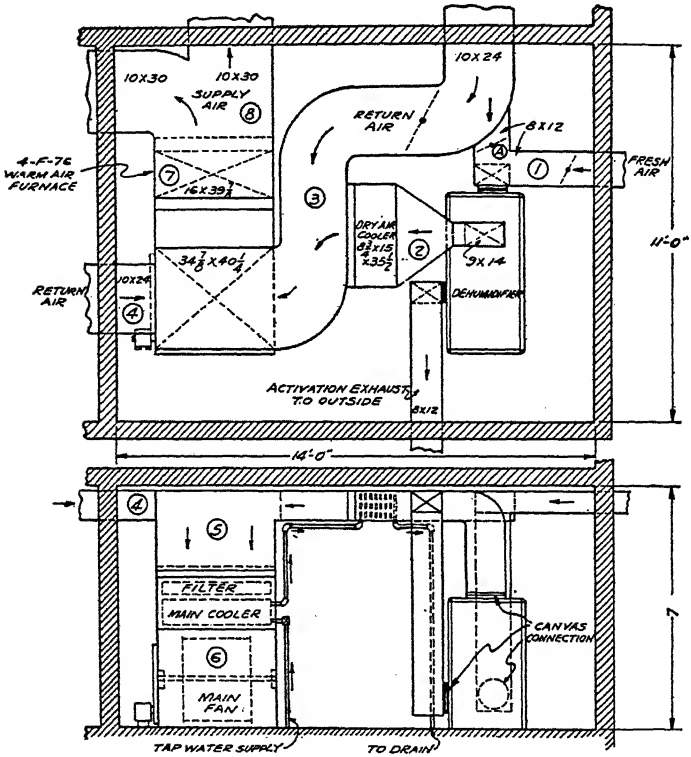


Fig. 95. Silica Gel Method of Dehumidif[illegible], Showing Typical Layout of Dehumidifier with Main Fan, Main [illegible]ler, and Warm-Air Furnace
Courtesy of The Bryant Heater Co.

Note: Manufacturers' catalogues and data should be secured because of improvements constantly being made.

doors and discharges the warm air through attic windows or louvers. Thus the cooler air is brought into the house and cools it to within a few degrees of the outside air. Such fans are selected, generally, in exactly the same manner as for furnace blowers. This system lacks filters and therefore has the disadvantage of drawing in dust, pollen, etc. to an extent that becomes undesirable.

The following facts were found by actual tests at the University of Illinois experimental station.

1. An uninsulated building of ordinary residential type may require the equivalent of three tons of ice in 24 hours on days when the maximum outdoor temperature reaches 100°F. if an effective temperature of approximately 72 degrees is maintained indoors.

2. The use of awnings at all windows in east, south, and west exposures may result in savings of from 20 to 30 per cent in the required cooling load.

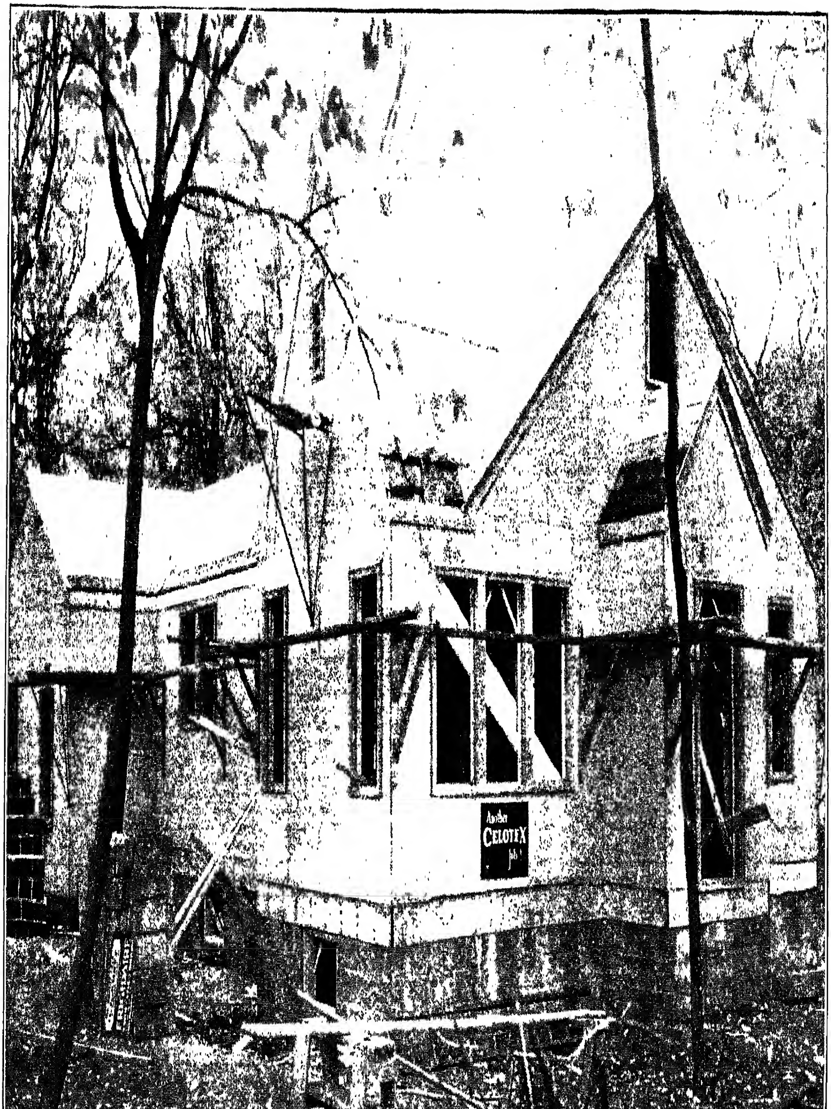
3. The cooling load per degree difference in temperature is not constant but increases as the outdoor temperature increases.

4. The heat lag of the building complicates the estimation of the cooling load under any specified conditions and makes such estimates, based on the usual methods of computation, of doubtful value.*

5. The seasonal cooling requirements are extremely variable from year to year, and the ratio between the degree-hours of any two seasons occurring within a 10-year period may be as high as 7.5 to 1. Hence an average value of the degree-hours cooling per season is comparatively meaningless.

6. The results of the tests suggest the use of a fan at night either to provide more comfortable conditions during the following day without provision for cooling, or to reduce the load required for cooling during the following day.

*Heat lag is explained in detail in Chapter V, Vol. II.



TYPICAL WALL INSULATION

Courtesy of Celotex Company

CHAPTER VIII

REGISTERS AND GRILLES

Registers and grilles are made of cast iron, bronze, pressed steel, and wood in many shapes and patterns. They are finished in practically every conceivable color and surface to meet the de-

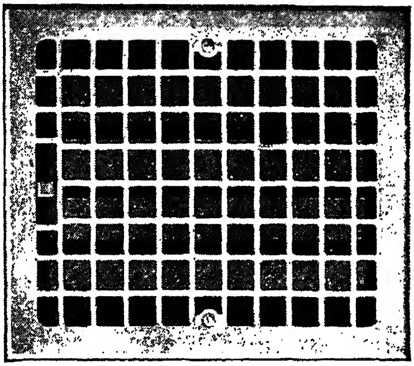


Fig. 96. Cast Floor Type of Hot-Air Register
Courtesy of The Auer Register Company

mands of interior decoration and design. The common types, considering styles, application, and design, are as follows.

Hot Air Furnace Registers. Registers used in connection with a hot air furnace are of the following four types: floor, baseboard, wall, and cold air.

Floor Type. Fig. 96 shows the most common type of floor register used in connection with hot air furnaces. This register has a cast metal face, with a bevel edge at the margin so it can be screwed down to the flooring, a steel body, and louvers. The louvers control direction of heat flow and the opening or closing of the register. Table 68 gives the size of opening, for the various common sizes, air opening of register, and the size for pipe or leader.

Baseboard Type. Baseboard registers, as shown in Fig. 97, are upright in position and are designed to be used in a wall at the floor level so that the baseboard frames it on either side. Table 69 gives

typical sizes and auxiliary dimensions, including area and pipe sizes for baseboard registers.

Table 68. Typical Sizes and Auxiliary Dimensions for Floor Registers

Size of Opening	For Size Pipe	Air Opening of Register	Size of Opening	For Size Pipe	Air Opening of Register	Size of Opening	For Size Pipe	Air Opening of Register
6x 8	6″	30″	10x14	10″	95″	14x18	15″	176″
8x 8	7″	45″	12x12	10″	97″	16x20	16″	210″
8x10	8″	54″	12x14	12″	115″	18x20	18″	250″
8x12	8″	64″	12x15	12″	130″	20x24	20″	330″
9x12	9″	76″	12x18	14″	150″			
10x12	10″	84″	14x16	14″	156″			

Table 69. Typical Sizes and Auxiliary Dimensions for Baseboard Registers

Register Size	Depth of Side Flange	Face Net Area	For Size Pipe	Area of Pipe	Register Size	Depth of Side Flange	Face Net Area	For Size Pipe	Area of Pipe
8x10	2¼″	58″	8″	50″	11x13	4″	115″	12″	113″
8x12	2¼″	70″	8″	50″	11x13	5¼″	115″	12″	113″
9x12	2¼″	80″	9″	64″	12x14	5¼″	126″	12″	113″
9x12	3¼″	80″	10″	78″	14x14	5½″	154″	14″	154″
10x12	3¼″	90″	10″	78″					

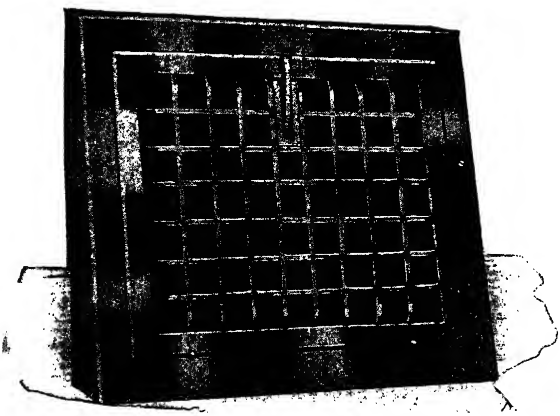


Fig. 97. Baseboard Type of Hot-Air Register
Courtesy of The Auer Register Company

Cold Air Type. Fig. 98 shows a typical design of a cold air steel register. Other types are constructed of wood or a combination of wood and steel. These registers, because of their large size and location in the floor, must be constructed to withstand practically the same stresses as the floor. Table 70 shows typical sizes with auxiliary

dimensions and pipe sizes commonly used. The net area of cold air register should be given special care when selections of sizes are made to insure ample free area.

Table 70. Typical Sizes and Auxiliary Dimensions for Cold Air Registers

Size of Registers	For Size Pipe	Area of Pipe	Face Air Opening	Size of Registers	For Size Pipe	Area of Pipe	Face Air Opening
12x14	12″	113″	124″	16x24	20″	314″	288″
14x16	14″	154″	168″	6x30	13″	132″	136″
6x24	12″	113″	108″	8x30	14″	154″	177″
8x24	14″	154″	144″	10x30	16″	201″	221″
10x24	14″	154″	177″	12x30	18″	254″	265″
12x24	16″	201″	212″	14x30	20″	314″	315″
14x24	18″	254″	254″				

Table 71. Typical Sizes and Auxiliary Dimensions for Out-of-Wall Registers

Register Size	Depth of Side Flange	For Size Pipe
12x9	6¼″	9″
12x10	7¼″	10″
13x11	8¼″	12″

Fig. 98. Cold-Air Type Register
Courtesy of The Auer Register Company

Out-of-Wall Registers. Fig. 99 shows a special type of register used only when a baseboard register cannot be set in the partition, because of structural obstructions. This condition should be avoided, if possible, because the protruding register is somewhat of an eye sore to an enclosure and makes the use of the surrounding area restricted. Table 71 shows typical sizes and other auxiliary dimensions for out-of-wall registers.

Register Boxes. A register box is the connecting link between a register and a leader, or stack. A register is generally larger in gross area than its supply leader, or stack, and therefore requires some

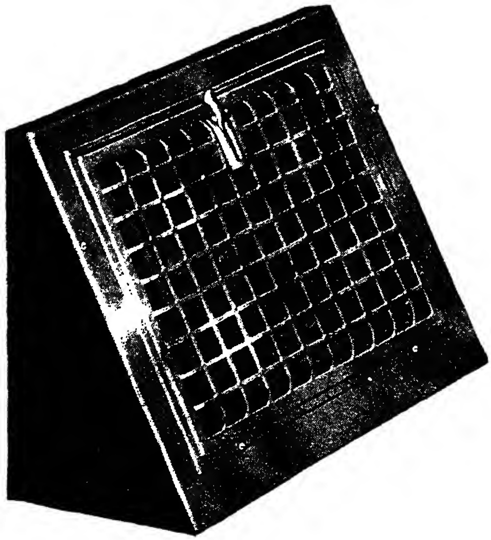


Fig. 99. Out-of-Wall Type Register
Courtesy of The Auer Register Company

means of connection. Also the leader, or stack, is round or rectangular in cross section and requires the use of a transition piece.

Floor registers are not permitted unless for some special purpose,

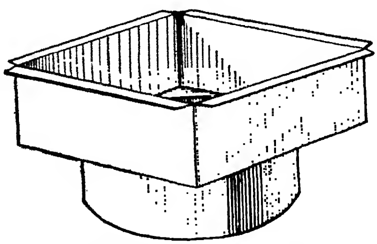


Fig. 100. Floor Register Box

as in entrance halls to dry shoes and to heat the hall. In case such registers are used, however, suitable boxes, Fig. 100, must be provided. The registers are preferably constructed with double walls

and tapered from the round collar to the top of the rectangular box. Because of the infrequent use of such registers, no standard sizes are shown here.

Baseboard registers have boxes like the ones shown in Figs. 101 and 102 and Figs. 103 and 104 for first and second floors respectively.

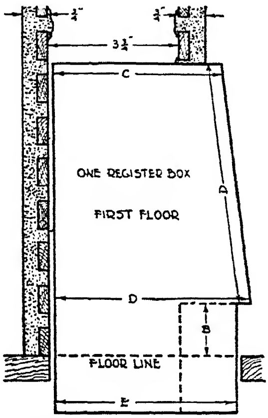


Fig. 101. Register Box for First Floor Baseboard Registers

Depth of Flange	Width A	B	C	D	E
2¼	10 x 8	2⅛	5¼	6½	6⅛
2¼	12 x 8	2⅛	5¼	6½	6⅛
2¼	12 x 9	2⅛	5¼	6½	6⅛
3¼	12 x 9	2⅛	5½	7¼	7⅛
3¼	12 x 10	2⅛	5½	7¼	7⅛
3¼	13 x 10	2⅛	5½	7¼	7⅛
4	13 x 11	2⅛	5½	8	7⅞
5¼	13 x 11	2⅛	5½	9¼	9⅛
5¼	14 x 12	2⅛	5½	9¼	9⅛
5½	14 x 13	2⅛	5½	9½	9⅜
5½	14 x 14	2⅛	5½	9½	9⅜

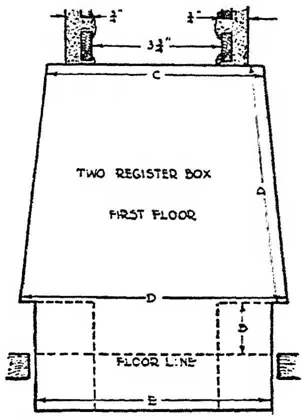


Fig. 102. Double Register Box for First Floor Baseboard Registers

Depth of Flange	Width A	B	C	D	E
2¼	10 x 8	2⅛	6¾	9¼	9
2¼	12 x 8	2⅛	6¾	9¼	9
2¼	12 x 9	2⅛	6¾	9¼	9
3¼	12 x 9	2⅛	7¼	11¼	10¾
3¼	12 x 10	2⅛	7¼	11¼	10¾
3¼	13 x 10	2⅛	7¼	11¼	10¾
4	13 x 11	2⅛	7¼	12¾	12¼
5¼	13 x 11	2⅛	7¼	15¼	14½
5¼	14 x 12	2⅛	7¼	15¼	14½
5½	14 x 13	2⅛	7¼	15½	15
5½	14 x 14	2⅛	7¼	15½	15

These boxes are made in such typical sizes as shown in Figs. 101 and 102 and Figs. 103 and 104. The dimensions shown for *E* in Figs. 101, 102, 103, and 104 can be changed to fit the size of wall pipe if necessary. Also if thickness of the wall should be materially different from what is shown, boxes at *C* and *D* must be changed accordingly. The standard and typical sizes shown in Figs. 101, 102, 103, and 104 are made to be assembled with registers like the type shown in Fig. 97.

Cold air registers do not require boxes as explained for hot air types. Instead, the pipe is connected to a transition piece or shoe, as

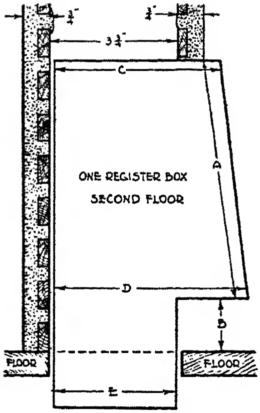


Fig. 103. Register Box for Second Floor Baseboard Registers

If studding and plaster are less than indicated, C and D must be reduced accordingly.

Depth of Flange	Width A	B	C	D	E
2¼	10 x 8	2⅛	5¼	6½	3⅝
2¼	12 x 8	2⅛	5¼	6½	3⅝
2¼	12 x 9	2⅛	5¼	6½	3⅝

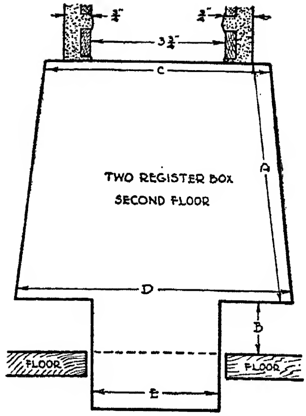


Fig. 104. Double Register Box for Second Floor Baseboard Registers

If thickness of wall is materially more or less than shown, change C and D accordingly.

Depth of Flange	Width A	B	C	D	E
2¼	10 x 8	2⅛	6¾	9¼	3⅝
2¼	12 x 8	2⅛	6¾	9¼	3⅝
2¼	12 x 9	2⅛	6¾	9¼	3⅝

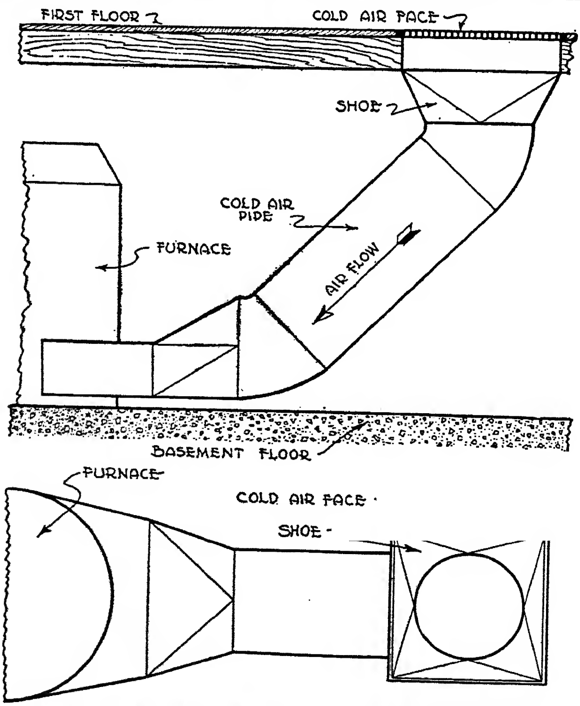


Fig. 105. Plan and Elevation of Recirculating Duct and Shoe

shown in Fig. 105 and this shoe forms a box similar to a hot air register box.

Out-of-wall registers employ a tapered shoe, see Fig. 106, as a box. This serves merely as a transition piece between the round leader and the rectangular shape of the register. The piece shown is a typical shape.

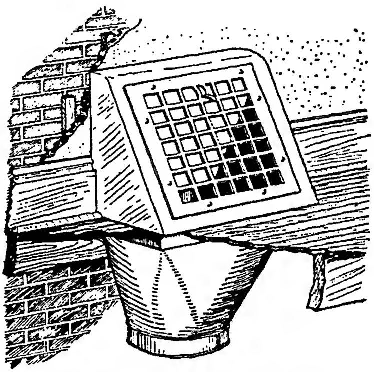


Fig. 106. Out-of-Wall Register Box

Air-Conditioning Grilles. Grilles used in connection with air-conditioning systems are constructed much smaller than hot air registers, because of forced air flow rather than gravity air flow. They are installed at points generally above the floor at the base-board levels. Grilles are usually more ornate than registers and the direction of air flow through them is more accurately controlled, which is important where forced air is employed.

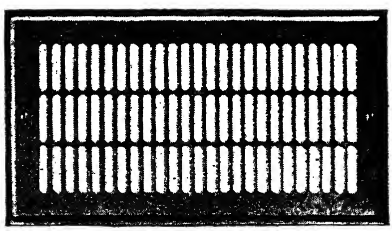


Fig. 107. Baseboard Grille
Courtesy of The Auer Register Company

Baseboard Grilles. This type of grille, see Fig. 107, is set directly in the baseboard. While the limited height of a baseboard allows only

small vertical dimensions, the length of the grille may be made as long as necessary to give the required grille surface area. Baseboard grilles are generally used for return air flow. They fit the top of the stack by a simple flange arrangement as shown in Figs. 108 and 109.

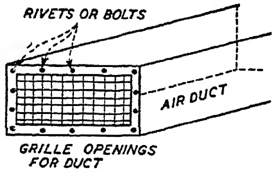


Fig. 108. Grille Air Duct Opening Installed on Air Duct

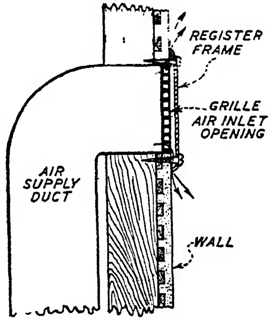


Fig. 109. Grille Air Duct Openings Installed on Wall Surfaces or F boards

Wall Grilles. Fig. 110 shows a typical wall type grille. Wall grilles can be obtained in many different patterns and are constructed with or without shut-off valves, air deflectors, and sliding arrangements for exact positioning between studs. Wall grilles are secured to the duct, as explained for baseboard grilles, or they are mounted on a steel frame, which is attached to the studs.

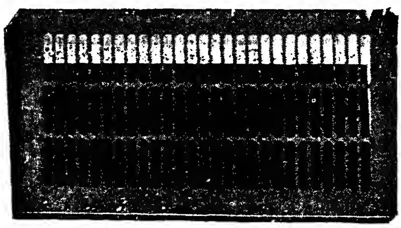


Fig. 110. Wall Type Grille
Courtesy of The Auer Register Company

Capacities and Sizes of Grilles. Table 72 gives a partial list of capacities and sizes for typical grilles at various air velocities. From these capacities it can be seen that limited vertical dimensions are the rule and that horizontal dimensions are much longer than the heights. Table 72 applies to both baseboard and wall grilles. The

4-, 5-, and 6-inch dimensions refer to baseboard grilles, while 6-, 8-, and 10-inch dimensions refer to wall grilles.

Design of Registers and Grilles. There are tables which indicate what size register to use with a particular pipe size for most of the ordinary registers, like the ones shown in this section. For instance, Fig. 97 shows a baseboard register and Table 69 shows the register sizes. For a pipe 9 inches in diameter, a register 9x12 inches should be used. Table 69 applies only to the particular register shown in Fig. 97. However, for all other registers or for all other pipe sizes, the register size should be such that the free area is at least equal in area to the leader supplying it. If possible, the register should be approximately the same width as the stack to which it is attached.

If a table, such as Table 69 is not available for a particular type of register, the above rule may be followed.

The size of cold air ducts or return pipes is discussed at a previous point in the section on "Furnace Heating," also the sizes of grilles or registers for mechanical warm-air systems are specified in the Technical Code found in the same section.

Table 72. Capacities and Sizes of Grilles

Register Stackhead Size	Open (Free) Area Sq.In.	Cubic Feet Per Minute Delivered through Register at Following Velocities:				
		200	250	300	380	400
8x 6	26	37	46	55	70	75
10x 4	22	30	38	46	58	61
10x 5	28	39	49	59	75	78
10x 6	36	50	62	74	94	100
10x 8	50	69	86	104	132	139
12x 4	28	39	49	59	75	78
12x 5	38	53	66	79	101	106
12x 6	43	60	75	91	115	121
12x 8	61	84	106	127	161	169
12x 9	69	95	120	143	182	192
12x10	78	108	135	163	206	217
14x 4	30	42	52	63	80	84
14x 5	41	57	71	85	108	113
14x 6	50	69	86	104	132	140
14x 8	71	99	124	149	188	198
14x10	92	127	159	191	243	256
16x 4	35	48	60	72	91	96
16x 5	47	65	82	98	124	131
16x 6	58	80	100	120	153	161
16x 8	82	114	143	172	218	229
16x10	106	147	184	221	280	294
18x 4	38	53	66	79	101	106
18x 5	53	73	91	110	139	147
18x 6	65	90	112	135	171	180
18x 8	92	127	159	191	243	256
20x 4	44	61	76	91	116	122
20x 5	59	82	103	123	156	165

Register Stackhead Size	Open Free Area Sq.In.	Cubic Feet Per Minute Delivered through Register at Following Velocities:				
		200	250	300	380	400
20x 6	75	104	130	156	197	208
20x 8	104	144	180	217	275	289
20x10	130	181	226	271	344	362
22x 4	48	66	83	100	126	133
22x 6	82	114	143	172	218	229
24x 4	53	73	91	110	139	147
24x 5	71	99	124	149	188	198
24x 6	90	125	156	187	237	250
24x 8	125	174	217	261	331	348
24x10	160	223	279	334	424	446
26x 4	57	79	99	118	150	158
26x 6	97	135	168	202	256	270
28x 4	61	84	106	127	161	169
28x 5	83	115	144	173	219	231
28x 6	105	146	182	219	277	292
28x 8	147	204	255	306	388	408
30x 4	66	92	115	138	175	185
30x 5	89	123	154	185	235	247
30x 6	113	157	196	236	298	314
30x 8	158	220	274	329	417	439
30x10	202	281	351	422	534	562
36x 4	79	110	137	164	208	219
36x 5	106	147	184	221	280	294
36x 6	136	189	236	283	359	378
36x 8	189	262	328	394	498	525
36x10	242	336	420	504	638	672

UNIT HEATERS AS UTILIZED IN A MANUFACTURING PLANT

Courtesy of McMay, Incorporated

CHAPTER IX

ELECTRIC HEATING

Until comparatively recent times, the cost of electric heating, for all purposes and under most conditions, has been prohibitive because of high power rates and poorly designed heating equipment. However, with power rates for heating gradually lowering

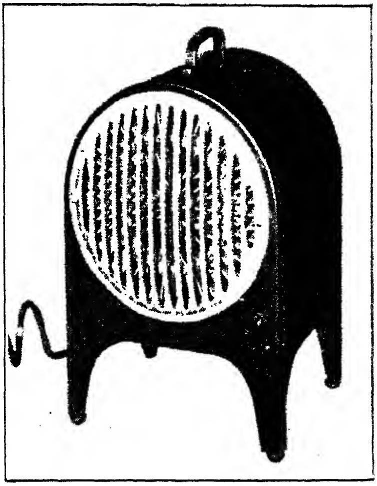


Fig. 111. Portable Electric Heater with Grille
Courtesy of Electric Air Heater Company, Mishawaka, Indiana

in many localities, and with the introduction of efficient electric heaters of sufficient capacity to heat any room or building rapidly, partial or major electric heating has become more economical and satisfactory. Electric heat has several advantages. It is clean, instantly available, easy to regulate, requires little space for equipment, and no space at all for fuel storage. In addition, the equipment can be moved easily to satisfy possible varying needs. If the heater is of the portable type, as shown in Fig. 111, it can be placed in any convenient location because it does not require pipes, ducts, or other accessories, common to other forms of heating.

Healthful Qualities of Electric Heat. To understand the value of electric heat from the standpoint of health, it should be remembered that the air which is breathed is made up of various component parts and that air is at its best when the chemical relationship between such component parts is not changed. To remove or alter any of these component parts produces an air that is unnatural and, therefore, not the best for breathing. Wherever combustion takes place, which is the case in all ordinary heating, oxygen is removed from the air by the combustion. Electric heaters require no combustion process and, as merely heating the air does not change its chemical relationship, it can be seen easily that electric heaters have this advantage over some of the other types of heaters.

Feasible Conditions for Electric Heating. With cheap power (approximately $0.02 per kilowatt-hour) electric heating could be employed economically in any part of the United States regardless of temperatures or the length of the heating season. On the other hand, higher rates confine electric heating to special localities and to special purposes.

In northern localities, electric heating is generally not economical as a major heater, except at those places where cheaper power is available. Therefore, its use is confined to such special seasons as spring and fall, during intervals when maximum heat is required, and as a temporary heat source when the regular heating system is shut down for repairs, etc.

In early spring and early fall, there are apt to be occasional cool days or evenings when some heat would make for comfort. It would not be practical, however, to start a heating system for such short periods. At such times, electric heaters prove economical and provide the required comfort. Where other types of heating are used as the major source of heat, the electric heater can aptly serve as an auxiliary supply during especially severe cold periods or at times when peak loads are required. The electric heater can also serve as a safety factor for possible periods, when for some cause the major heating system has to be shut down.

In southern or southwestern parts of the United States where mild winters prevail, the electric heater can be used economically regardless of power rates because heat is required mainly for short periods, generally morning and evening.

Another condition under which electric heating can be used to good advantage is in factories, or similar places, where a large minimum load charge is made for electric power, due to the large size of connected motors. In such cases, the amount of electric power which must be paid for is not actually used. Thus, electric heaters could be used not only economically but practically free of charge in such instances.

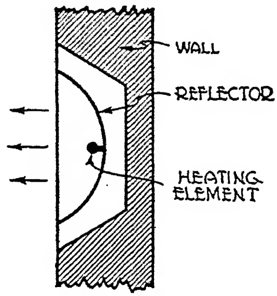


Fig. 112. Radiant Electric Heater

There are many miscellaneous conditions where electric heating proves not only economical, but it also provides comfort and convenience in any climate where heat is required. In large commercial buildings it is not economical to keep the major heating system in operation over Sunday. However, in many instances one or more offices in such a building might need heat. This could be handled by small electric heaters. Also, submarines, press boxes, stadiums, watchman towers, signal towers, pump or valve stations, isolated rooms, garages, dressing rooms, temporary rooms or construction houses, airplane hangars, etc., could all be supplied with electric heat on a sound basis in any climate.

Principles of Electric Heating. Where electricity is used to heat the air, either radiant or convection principles are employed. In the radiant system, Fig. 112, the heater is recessed in a wall, or may be mounted on a pedestal. The heating element provides the heat and the reflector radiates the heat outward in the direction of the arrows. This type of heater has limitations due to the fact that it warms only the air directly in front of it and not the whole room or enclosure.

Heaters using convection principles are made in two types, gravity and fan. The gravity type can be used as shown in Fig. 113, or recessed as shown in Fig. 114. In either case, air enters at

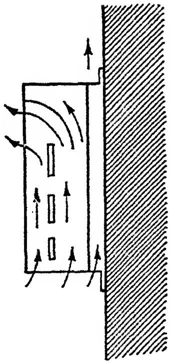


Fig. 113. Gravity Convection Wall Type Heater

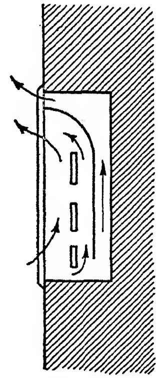


Fig. 114. Recessed Gravity Convection Wall Type Heater

the bottom of the heater. It is heated by the elements, and discharged through the top.

The fan type heater employs an electric fan, in addition to the elements in the gravity system, to circulate the warm air. Today the fan type of electric heater is, for most conditions, the most

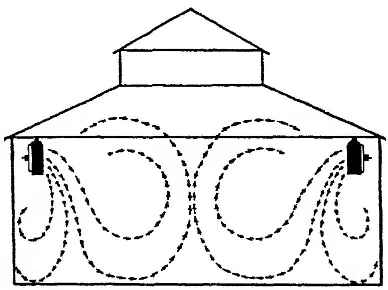


Fig. 115. Circulation Brought About by Fan Type Heaters

efficient because it produces forced circulation of warm air, thus providing more uniform and positive room temperatures. The circulation brought about by the fan type can be illustrated by Fig. 115. While this figure shows the effect of the suspended type of heater,

the principle is the same for a heater located elsewhere in a room or enclosure. The deflectors can be adjusted so as to cause the warm air coming from the heater to be deflected up, down, or sideways.

Portable Electric Heaters. The portable type electric heater, Fig. 111, was designed for portable use in homes, offices, etc. where

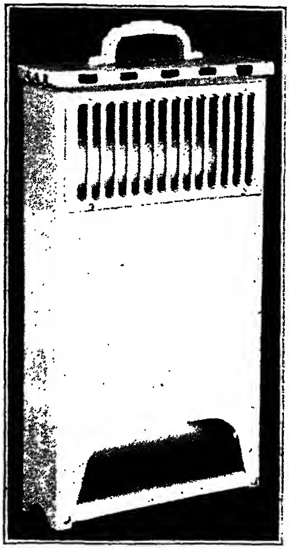


Fig. 116. Portable Gravity Convection Heat
Courtesy of Electric Air Heater Company, Mishawaka, Indiana

occasional heat might be needed. A round electric heating unit (described on page 202) is used in this heater. A fan and motor are mounted behind the heating unit, giving forced air circulation. The fan blades are pitched and balanced to produce quiet operation and fixed temperature. The motors are of the shaded pole type.

Fig. 116 shows a portable gravity electric heater which operates using the same type of heating unit as Fig. 111 except that it depends on gravity instead of a fan for the circulation of air.

Fig. 117 shows the main features of a fan-type portable electric heater. The air is sucked in from the rear side of the heater and then forced by the fan, through the heating unit out into the room. Table 73 gives the dimensions and capacity ratings for various sizes

***Table 73. Dimensions and Capacities of Various Sizes of Fan-Type Portable Electric Heaters**

Model	†Dimensions, Inches						Face Area Sq. Ft.	Diam. Fan
	A	B	C	D	E	F		
A-15 A-20	13½	8¾	4¾	8⅝	9⅝	3½	.327	7¾
B-30 B-40	15½	10	5½	8⅞	11	4	.442	9
C-50 C-60	18⅜	11½	6⅞	10½	13¾	4¼	.60	11½

Capacity Ratings

Model	K.w.	B.t.u.	E.d.r.	60° Temp. Rise		45° Temp. Rise	
				C.f.m.	F.p.m.	C.f.m.	F.p.m.
A-15 A-20	1.5 2	5122 6830	21.3 28.4	75 100	220 305	103 138	310 420
B-30 B-40	3 4	10245 13660	42.7 56.9	150 200	340 450	207 276	468 625
C-50 C-60	5 6	17075 20490	71.1 85.3	260 310	435 515	345 415	575 692

*Courtesy of Electric Air Heater Company, Mishawaka, Indiana.
†Letters refer to dimensions shown in Fig. 117.

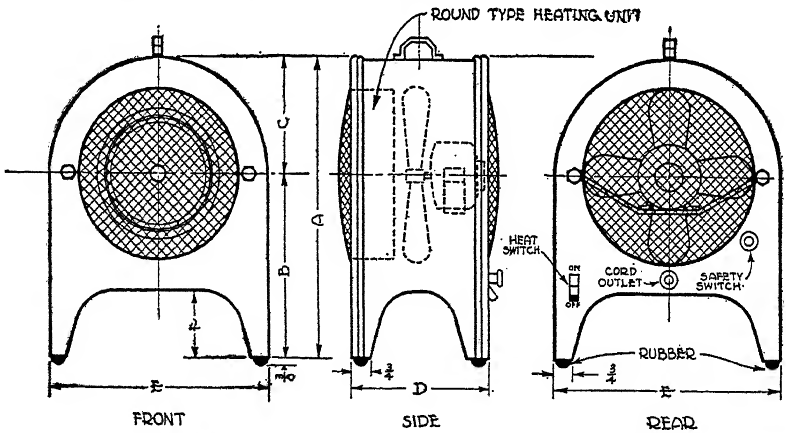


Fig. 117. Working Drawings Showing Construction of a Fan Type Portable Electric Heater
Courtesy of Electric Air Heater Company, Mishawaka, Indiana

of such heaters. From the table it can be seen that to supply 10,245 B.t.u. per hour, or approximately that amount, it would be necessary to use a Model B-30 heater, and that the A, B, C, D, E, and F dimensions, as shown in Fig. 117, would be 15½, 10, 5½, 8⅞, 11, and 4 inches respectively. The determination of the B.t.u. re-

quirement is found by calculating the heat loss in B.t.u. for the room or enclosure in question. The calculation of heat loss is explained elsewhere in this book.

Wall-Type Electric Heaters. The wall-type heater, Fig. 118, has a wide range of application and can be used for complete elec-

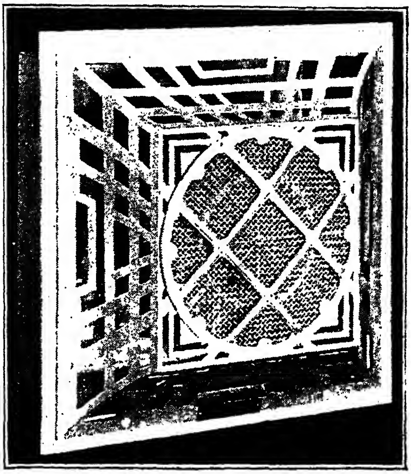


Fig. 118. Wall-Type Electric Heater
Courtesy of Electric Air Heater Company, Mishawaka, Indiana

tric heating systems or for auxiliary work. As used for complete or major heating, the wall-type heater is installed in either 4 or 6-inch walls. Fig. 119 shows details for the installation of a wall-type heater. An assembly consists of a grille, a round heating unit (see

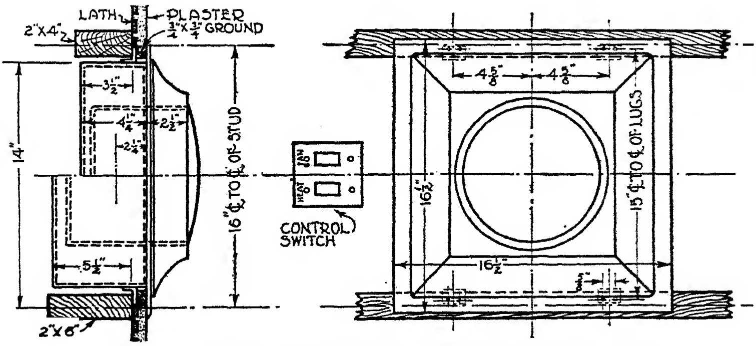


Fig. 119. Installation Drawings for Placing Wall-Type Electric Heaters

Fig. 120), a fan, a motor, and a metal box which fits into the wall. It can be controlled by either a manual or a thermostatic switch. These heaters are installed as near to the floor as possible above the baseboard in any residence room and in any type enclosure.

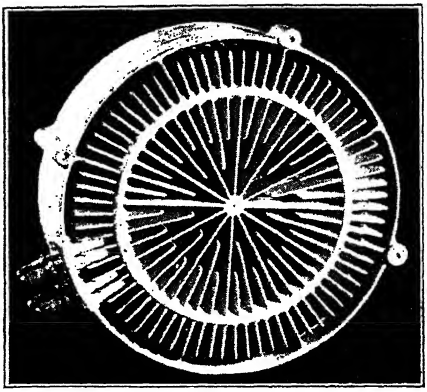


Fig. 120. Circular Heating Unit
Courtesy of Electric Air Heater Company, Mishawaka, Indiana

These heaters are especially suitable for residence work and for auxiliary heating. Table 74 shows the capacity ratings including B.t.u. output. The selection of a model depends on the B.t.u. requirement.

Industrial Electric Heaters. Fig. 121 shows an electric heater which is used for permanent industrial applications. Such heaters are suspended from ceilings or from wall brackets. In keeping

***Table 74. Capacity Ratings for Wall-Type Electric Heaters**

Model Studs 2x4	Model Studs 2x6	Volts	Kw.	B.t.u.	E.d.r.	60° F. Temp. Rise C.f.m.	60° F. Temp. Rise F.p.m.	Diam. Fan
AA1010-4	AA1010-6	110	1.0	3415	14.2	50	175	7
AA1020-4	AA1020-6	220						
AA1510-4	AA1510-6	110	1.5	5122	21.3	75	260	7
AA1520-4	AA1520-6	220						
AA2010-4	AA2010-6	110	2.	6830	28.4	100	350	7
AA2020-4	AA2020-6	220						
AN3010-4	AN3010-6	110	3.	10245	42.7	150	405	8
AN3020-4	AN3020-6	220						
	BN5020-6	220	5.	17075	71.1	260	542	9¼
	BN6020-6	220	6.	20490	85.3	310	646	9¼

1 Kw. Hr. = 3415 B.t.u. = 14.23 sq. ft. Equivalent in Direct Radiation. 60 cycle A.C. is Standard.

*Courtesy of Electric Air Heater Company, Mishawaka, Indiana.

with the general principles thus far explained, of creating forced circulation, this heater has a fan placed behind the heating unit. This type of heater is used extensively in factories, warehouses, and other large enclosures.

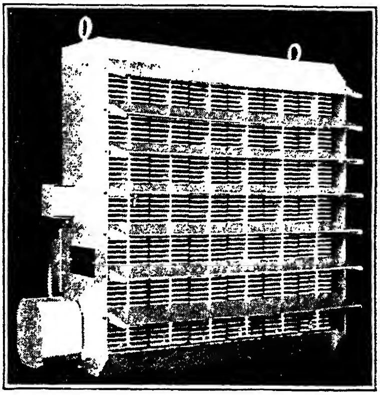


Fig. 121. Industrial Electric Heater
Courtesy of Electric Air Heater Company, Mishawaka, Indiana

Table 75 shows capacity ratings for various industrial heaters and, like previously mentioned types, they are selected after determining the total B.t.u. requirements for the enclosure.

***Table 75. Capacity Ratings for Industrial Electric Heaters**

Models	Ratings			60°F. Temp. Rise		45°F. Temp. Rise		Fan Diam.
	Kw.	B.t.u.	E.d.r.	C.f.m.	F.p.m.	C.f.m.	F.p.m.	
8-2	2	6830	28.5	100	333	135	450	6½
8-3	3	10245	42.7	155	515	207	690	
11-4½	4.5	15367	64	233	382	310	596	9
11-6	6	20490	85	310	508	412	660	
14-10	10	34150	142	517	488	684	645	12
14-15	15	51225	213	776	732	1034	975	
20-25	25	85375	356	1290	533	1720	712	18
20-35	35	119525	498	1800	750	2400	1060	
27-45	45	153675	640	2325	570	3066	748	24
27-60	60	204900	854	3100	760	4133	1000	
32-90	90	307350	1280	4656	727	6108	954	30
32-120	120	409800	1708	6208	968	8280	1294	
38-150	150	512250	2135	7760	785	10346	1118	36
38-180	180	614700	2572	9300	1000	12400	1340	

Heating Units. Fig. 120 shows a typical heating unit of circular or round construction. This type of unit is used in most heaters previously mentioned. The unit consists of a helical sheath

*Courtesy of Electric Air Heater Company, Mishawaka, Indiana.

wire type of resistance heater element, cast integral with aluminum fin-type grids.

Table 76 gives ratings for the various heating units.

**Table 76. Ratings for Circular or Round Type of Heating Units*

Model	Ratings				60°F. Temp. Rise	
Watts Volts	Kw.	Volts	B.t.u.	E.d.r.	C.f.m.	F.p.m.
A-1200-1	1.2	110	4098	17.1	62.6	190
A-1200-2	1.2	220	4098	17.1	62.6	190
A-1500-1	1.5	110	5122	21.3	75	220
A-1500-2	1.5	220	5122	21.3	75	220
A-2000-1	2.0	110	6830	28.4	100	305
A-2000-2	2.0	220	6830	28.4	100	305
B-2000-1	2.0	110	6830	28.4	100	226
B-2000-2	2.0	220	6830	28.4	100	226
B-3000-1	3.0	110	10245	42.7	150	340
B-3000-2	3.0	220	10245	42.7	150	340
B-4000-1	4.0	110	13660	56.9	200	450
B-4000-2	4.0	220	13660	56.9	200	450
C-4000-1	4.0	110	13660	56.9	200	277
C-4000-2	4.0	220	13660	56.9	200	277
C-5000-1	5.0	110	17075	71.1	260	435
C-5000-2	5.0	220	17075	71.1	260	435
C-6000-1	6.0	110	20490	85.3	310	515
C-6000-2	6.0	220	20490	85.3	310	515

1 Kw. Hr. =3415 B.t.u. =14.23 E.d.r.

Determining Heater Sizes. There are three common methods used to calculate heater sizes. The B.t.u. method is the most accu-

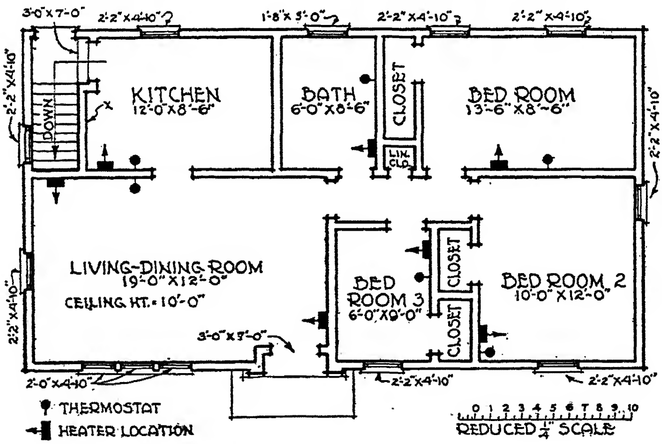


Fig. 122. Small House Floor Plan

rate in many ways, especially so, because with this method insulation can be considered. The "Watts Per Cubic Foot" method is approximate and no degree of accuracy could be expected from its use.

*Courtesy of Electric Air Heater Company, Mishawaka, Indiana.

The "35" method is fairly accurate for common construction types, unless insulation is used in the construction. It has been developed by constant trial over a number of years and, where exact results are not required, it is a quick design method.

In the following example, the B.t.u. and "35" methods are illustrated, using the same house and temperature conditions.

B.t.u. Method of Selecting Heaters. The following example is typical and the calculations clearly illustrate the method.

Example 1. It is required to make a heating analysis of the house plans shown in Fig. 122, to select the proper sizes of wall-type electric heaters, such as Fig. 118, to design the controls, and to select locations for the heater.

***Solution.** The following solution takes into account only the floor plan shown. The basement is not considered beyond making the assumption that the basement temperature is 32°F. The attic temperature is assumed to be the same as outside temperature, because of the ventilating openings. The solution follows the approved methods given in the section on "Transmission Coefficients and Tables" and "Heating and Cooling Loads" in Vol. II, Chapter VI.

The solution is based on the following conditions and construction:

(1) Location....................................El Paso, Texas
(2) Low outside temperature....................................-5°F.
(3) Base temperature....................................$=(-5°+15°)=+10$°F.
(4) Prevailing wind....................................11 m.p.h. (N.W.)†
(5) Inside temperature....................................70°F.
(6) Construction:
Walls: Frame with 2×4-inch studs with sheathing, and siding on the outside, wood lath and plaster on the inside.
Roof: Not considered.
Floors: Rough and finished flooring on 2×6-inch joists.
Doors: Two 3×7-foot wood doors, 1⅝ inches thick.
Windows: As in Fig. 122, Single, double-hung.
Ceiling: 1-inch Insulite plaster backing and plaster, on joists.
(7) Transmission Coefficients:
Walls: (Table 5) Vol. II, Chapter V....................................U = .25
Roof: Not considered....................................
Floors: (Table 8) Vol. II, Chapter V....................................U = .34
Doors: (Table 13B) Vol. II, Chapter V....................................U = .46
Windows: (Table 13A) Vol. II, Chapter V....................................U = 1.13
Ceiling: (Calculated)....................................U = .25
(8) Infiltration Coefficients:
Windows: Average window, non-weatherstripped, $\frac{1}{16}$-inch crack and $\frac{3}{64}$-inch clearance. The leakage per foot of crack for an 11-mile wind velocity is 25 c.f.h. This was determined by interpolation of Table 19 in Vol. II, Chapter V. The heat equivalent per hour per degree per foot of crack is taken from Vol. II, Chapter V.
$25\times 0.018=0.45$ B.t.u. per degree Fahrenheit per foot of crack

*Data Courtesy of Electric Air Heater Company, Mishawaka, Indiana. †Table 25

Doors: Assume infiltration loss through door cracks twice that of windows or

$2 \times 0.45 = 0.90$ B.t.u. per degree Fahrenheit per foot of crack

Walls: A plastered wall allows so little infiltration that in this problem it may be neglected.

(9) Calculations: Table 77 shows the calculations for the various items, such as net exposed wall area in square feet, ceiling area in square feet, etc.

The living room is considered first, as shown in Table 77. The plans, Fig. 122, show that the living room is 19×12 feet. Two walls are exposed. These are the walls of the living room through which transmission losses occur.

Note: If the reader has not become familiar with the section on "Transmission Coefficients and Tables," he is advised to do that before going further.

The area of the walls is figured by multiplying length by ceiling height. Thus the 19-foot wall has an area of $19 \times 10 = 190$ square feet. Also the 12-foot wall has an area of $12 \times 10 = 120$ square feet. The gross area of the two walls is therefore $190 + 120 = 310$ square feet. To obtain *net* area, subtract window area plus door area from the wall area.

The door area is 3×7 feet $= 21$ square feet.

The three windows in the 19-foot side are each $2'0'' \times 4'10''$. The area of one window is $4'10'' \times 2'0'' = 8$ square feet (approximately). Then the three windows have a combined area of $3 \times 8 = 24$. This is called 25 square feet to make up for the approximation. The window in the 12-foot side is $2'2'' \times 4'10''$ or $2\frac{1}{6} \times 4\frac{5}{6}$ feet $= 10$ square feet (approximately). The combined glass area is therefore

$$25 + 10 = 35 \text{ square feet}$$

The *net* wall area is then

$$310 - 35 - 21 = 310 - 56 = 254 \text{ square feet}$$

The *net* wall area, glass, and door areas can now be inserted in Table 77 in Column 1.

The living room floor has an area of

$$19 \times 12 \text{ feet} = 228 \text{ square feet}$$

This is put in Column 1 of Table 77.

The ceiling area is the same as the floor area. This is put in Column 2 of Table 77.

The glass crack is figured by adding the perimeters as illustrated in Fig. 123. The vertical lines *A*, *B*, *C*, and *D* represent the sides of the windows. The horizontal lines marked *E*, *F*, *G*, *H*, *I*, *J*, *K*, *L*, and *M* represent the top, meeting rail, and bottom of the windows. The total window crack is therefore the sum of lines *A* to *M*, or,

$$\begin{aligned} 4 \times 4'10'' &= 19'4'' \\ 9 \times 2'\ 0'' &= \underline{18'0''} \\ \text{Total} \quad &= 37'4'' \end{aligned}$$

The other window in the living room is $2'2'' \times 4'10''$. Its perimeter including meeting rail is

$$\begin{aligned} 2 \times 4'10'' &= \ 9'\ 8'' \\ 3 \times 2'\ 2'' &= \underline{\ 6'\ 6''} \\ \text{Total} \quad &= 15'14'' = 16'2'' \end{aligned}$$

Total window crack is $37'4'' + 16'2'' = 53'6''$

This is put in Column 3 of Table 77.

The door crack is figured by finding the perimeter of the door.

$$2 \times 7'0'' = 14'0''$$
$$2 \times 3'0'' = \underline{\ 6'0''}$$
$$\text{Total} = 20'0''$$

This is put in Column 4 of Table 77.

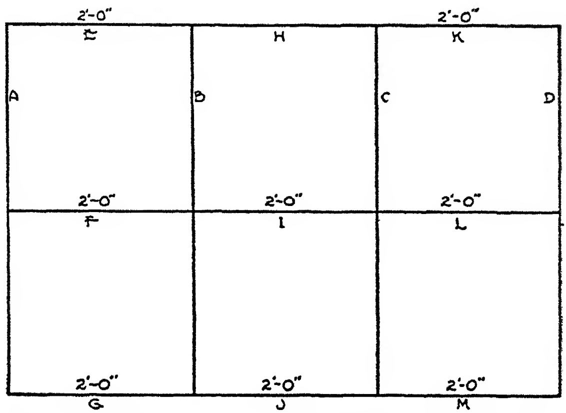


Fig. 123. Illustration of How to Calculate Cracks for Three Windows

The U values are found as explained in Items 7 and 8 and can be put in Column 5 of Table 77.

Temperature difference $(t-t_o)$ is found by figuring the difference between inside and outside temperatures. The inside temperature is 70°F. and the base temperature is 10°F., therefore, the temperature difference is 60°F. This can be put in Column 6 of Table 77.

Note: The attic temperature is considered as +10°F. so $(t-t_o)$ for ceiling is also 60°F. Basement is 32°F. so $(t-t_o)$ for floor is 38°F.

To calculate loss in B.t.u. per hour, Formula (1), Vol. II, is used.

Formula (1),

$$H_t = AU(t-t_o)$$

Substituting values in the formula for the wall of living room,

$$H_t = 254 \times .25 \times 60$$
$$H_t = 3{,}810 \text{ B.t.u.}$$

The loss in B.t.u. for floor, ceiling, glass, and door is found in the same manner as for the wall.

The loss for window cracks is taken as one-half the total amount. Use Formula (1), Vol. II, first.

$$H_t = 53.5 \times .45 \times 60 = 1{,}445$$

Then $\frac{1}{2}$ of $1{,}445 = 723$

This is put in Column 7 of Table 77.

The loss for door cracks is also taken as one-half of the total amount or ½ of 20 feet = 10 lineal feet. The same formula is applied as for glass crack and loss put in Column 7.

Table 77 presents the various calculations and losses for the other rooms of Fig. 122, which have been figured exactly as explained for the living room. The wall marked *X*, in the kitchen, is considered the same as an exposed wall. All closets and halls receive sufficient heat from the heated rooms. The kitchen

Table 77. Calculation Sheet Showing Basic Figures Used in Estimating Heat Losses of House in Fig. 122

Room	Net Exposed Area Sq. Ft.	Ceiling Area Sq. Ft.	Lineal Feet of Crack, Window	Lineal Feet of Crack, Door	*U* Value*	Temp. Diff.†	Loss in B.t.u. Per Hour‡
Living Room							
Wall	254	...	...	...	.25	60	3,810
Floor	228	...	...	...	.34	38	2,946
Ceiling	...	228	...	...	.25	60	3,420
Glass	35	...	...	...	1.13	60	2,373
Door	21	...	...	...	.46	60	580
Glass Crack	...	...	53½	...	.45	60	723‖
Door Crack	...	...	...	20	.90	60	540‖
Kitchen							
Wall	174	...	...	...	.25	60	2,610
Floor	102	...	...	...	.34	38	1,318
Ceiling	...	102	...	...	.25	60	1,530
Glass	10	...	...	...	1.13	60	678
Door	21	...	...	...	.46	60	580
Glass Crack	...	...	16	...	.45	60	216‖
Door Crack	...	...	...	20	.90	60	540‖
Bath							
Wall	55	...	...	..	.25	60	825
Floor	51	...	...	...	.34	38	659
Ceiling	...	51	...	...	.25	60	765
Glass	5	...	...	...	1.13	60	339
Door	...	...	...	...	...	...	...
Glass Crack	...	...	11	...	.45	60	149‖
Door Crack	...	...	...	...	...	...	...
Bedroom No. 1							
Wall	202	...	...	...	.25	60	3,030
Floor	114	...	...	...	.34	38	1,473
Ceiling	...	114	...	...	.25	60	1,710
Glass	18	...	...	...	1.13	60	1,220
Door	...	...	...	...	...	...	...
Glass Crack	...	...	32	...	.45	60	432‖
Door Crack	...	...	...	...	...	...	...
Bedroom No. 2							
Wall	202	...	...	...	.25	60	3,030
Floor	120	...	...	...	.34	38	1,550
Ceiling	...	120	...	...	.25	60	1,800
Glass	18	...	...	...	1.13	60	1,220
Door	...	...	...	...	...	...	...
Glass Crack	...	...	32	...	.45	60	432‖
Door Crack	...	...	...	...	...	...	...
Bedroom No. 3							
Wall	51	...	...	...	.25	60	765
Floor	54	...	...	...	.34	38	698
Ceiling	...	54	...	...	.25	60	810
Glass	9	...	...	...	1.13	60	610
Door	...	...	...	...	...	...	...
Glass Crack	...	...	16	...	.45	60	216‖
Door Crack	...	...	...	...	...	...	...

*Coefficients (*U*) found in or calculated by methods explained in Chapter V, Vol. II.

†Assume 70°F. inside temperature instead of mean temperature as in Illustrative Problem in Chapter VI, Vol. II. Assume basement temperature as 32°F.

‡All heat losses are calculated by Formula (1), Vol. II.

‖Assume one-half of window and door crack (See Vol. II, Chapter V).

door is considered the same as an outside door. It should be remembered that the assumed 32°F. basement temperature makes the temperature difference for the floor only 38°F. That is 70° − 32° = 38°F. The reader should understand that only walls, windows, and doors actually exposed to the outside, need to be considered in calculating heat losses. For example, in bedroom No. 1, the two outside walls are considered. The inside partitions are assumed as having 70°F. on both sides, so no heat loss takes place.

From Table 77 the heat losses in B.t.u. per hour for the various rooms are added and found to be as presented in Table 78.

Table 78. Total Heat Losses

Rooms	B.t.u. per Hour
Living Room	14,392
Kitchen	7,472
Bath	2,737
Bedroom No. 1	7,865
Bedroom No. 2	8,032
Bedroom No. 3	3,099

To find the size of heater in terms of kilowatts for each room, first refer to Table 74 and note that one kilowatt hour is equal to 3,415 B.t.u. So, to find the kilowatts needed for each room, divide the heat loss in B.t.u. by 3,415 as shown in Table 79.

Table 79. Conversion of B.t.u. to Kilowatts

Rooms	Heat Loss in B.t.u.	Required Kw.
Living Room	14,392 ÷ 3,415 =	4.21
Kitchen	7,472 ÷ 3,415 =	2.18
Bath	2,737 ÷ 3,415 =	.80
Bedroom No. 1	7,865 ÷ 3,415 =	2.30
Bedroom No. 2	8,032 ÷ 3,415 =	2.35
Bedroom No. 3	3,099 ÷ 3,415 =	.90

Table 74 gives capacity ratings for wall-type heaters. It should be noted that the first two columns indicate partition depth. The 2×4 means partitions or walls framed with 2×4-inch studs, while the 2×6 refers to 2×6-inch studs. The residence in Fig. 122 is framed using 2×4 studs, so only heaters appearing under the 2×4 heading may be considered.

In selecting heaters, it is the best practice to select a size slightly larger than necessary rather than too small.

The living room requires 4.21 kilowatts. There is not a heater of this rating or near it shown in Table 74 and the largest possible 2×4 heater is 3 kw. Therefore, one 3-kw. and one 1.5-kw. heaters are selected for the living room. Heaters for the other rooms are selected in like manner and are shown in Table 80.

Table 80. Selection of Suitable Heaters

Rooms	Required Kw.	Heater Size in Kw.
Living Room	4.21	Use one 3-kw. and one 1.5-kw.
Kitchen	2.18	Use one 3-kw.
Bath	.80	Use one 1-kw.
Bedroom No. 1	2.30	Use one 3-kw.
Bedroom No. 2	2.35	Use one 3-kw.
Bedroom No. 3	.90	Use one 1-kw.
Totals =	12.74 Kw.	15.5 Kw.

The entire heating system should be controlled by automatic thermostats. The degree of accuracy depends on economic considerations. (See Chapter XII on "Automatic Controls.")

Each room should have an individual control affecting only the heaters in that particular room. Any one of many systems of heat control can be employed, together with night and week-end shutoffs. The thermostats should be installed where the hot blast from the heaters does not strike them directly. See Fig. 122.

The heaters in the various rooms should be placed where the hot air from them will *wipe* the coldest wall surfaces. Fig. 122 shows recommended locations.

The "35" Method of Selecting Heaters. In Table 81 are shown various descriptions of structures and two approximate methods of determining the proper sizes for electric heaters. These methods are fairly accurate for structures of common construction and for temperatures ranging from 0°F. to 70°F. However, it should be kept in mind that where special or out of the ordinary conditions are encountered, the B.t.u. Method must be used for accurate results.

Table 81. Approximate Methods of Determining Electric Heater Sizes

The "Watts Per Cubic Foot" Method		The "35" Method	
Description	Watts per Cu. Ft.		
1. Interior room with little or no outside exposure	.75 to 1.25	1. Volume in cu. ft. for one air change × .35	equals watts
2. Average rooms with moderate windows and doors	1.25 to 1.75	2. Exposed net wall, roof or ceiling and floor in sq. ft. × 3.5	equals watts
3. Rooms with severe exposure and great window and door space	2.0 to 4.0	3. Area of exposed glass and doors in sq. ft. × 35.0	equals watts
4. Isolated rooms, cabins, watch houses and similar buildings	3.0 to 6.0		Total watts

The following example is a typical application of The "35" Method.

Example 2. It is required to make a heating analysis of the house shown in Fig. 122.

Solution. Items 1, 2, and 3 shown in The "35" Method in Table 81 are developed and shown in Table 82.

Table 82. Calculation Sheet Showing Basic Figures Used in "35" Method

Rooms	Item 1 of Table 81	Item 2 of Table 81	Item 3 of Table 81	Total Watts
Living Room	798	2,485	1,960	5,243
Kitchen	357	1,323	1,085	2,765
Bath	179	550	175	904
Bedroom No. 1	403	1,505	630	2,538
Bedroom No. 2	420	1,547	630	2,597
Bedroom No. 3	189	557	315	1,061

For the living room the dimensions are 19×12 feet and the room has a 10-foot ceiling. The volume of the room is therefore 19×12×10 feet = 2,280 cubic feet. Multiplying 2,280 by .35 as directed in Item 1 of Table 81, equals 2,280×.35 = 798. This is put in Column 1 of Table 82.

Item 2, of Table 81, shows that the net exposed wall, ceiling, and floor areas must be added and multiplied by 3.5. From Table 77 the figures 254 for net wall area, 228 for net floor area, and 228 for net ceiling area are taken. Added they equal 710 square feet. Then, 710×3.5 = 2,485. This is put in Column 2 of Table 82.

Item 3, of Table 81 shows that the area of exposed glass (windows) and door areas must be added and multiplied by 35.0. In Table 77 the figures 35 and 21, for glass and doors, are found. Added they equal 56. Then 56×35 = 1,960. This is put in Column 3 of Table 82. The same procedure is followed for the balance of the rooms.

The living room requires 798+2,485+1,960 = 5,243 watts. There are 1000 watts in 1 kilowatt. Therefore to find the kilowatt requirements of the living room, divide 5,243 by 1000 which equals approximately 5 kilowatts. This is close enough in an approximate method. The other rooms are handled in the same manner. Table 83 presents the conversion of watts to kilowatts.

Table 83. Conversion of Watts to Kilowatts

Rooms	No. of Watts	Approximate Kilowatts
Living Room	5,243 ÷ 1000 =	5
Kitchen	2,765 ÷ 1000 =	3
Bath	904 ÷ 1000 =	1
Bedroom No. 1	2,538 ÷ 1000 =	3
Bedroom No. 2	2,597 ÷ 1000 =	3
Bedroom No. 3	1,061 ÷ 1000 =	1
Total =		16 Kilowatts

Thus by The "35" Method 16 kilowatts would be required.

This compares favorably with the B.t.u. Method, by which the capacity of heaters selected, totaled 15.5 kilowatts.

Estimating Yearly Current Costs. The yearly, or heating season's, cost in terms of electrical energy, cannot be estimated accurately. No one can forecast the exact temperature and wind conditions that may prevail during the months when heating is required. The only bases that heating engineers can use with any degree of assurance are past records in cases where almost identical apparatus was used, and from these records construct an estimate. The following example is typical.

Example 3. Calculate the season's cost of operating the electric heating system calculated in Example 2. Assume in this case that the figures obtained apply to the Chicago area.

***Solution.** From tests and other reliable sources it is determined that the percentage of the season's heat requirements consumed during each month, is as shown in Table 84.

In Vol. II, Table 32 shows that for Chicago there are 6,315 degree-days† in the heating season. To make calculations easier, take an even 6,300.

Experience has shown that over a number of years electric heaters, such as those selected for Fig. 122, operating under average conditions, are *on* only approximately 10 per cent of the degree-day figure. This is based on rigid tem-

*Data Courtesy of Electric Air Heater Co., Mishawaka, Indiana.
†Explained in Vol. II, Chapter VI.

Table 84. Per Cent of Total Heat Required for Different Months of Heating Season

Months	Per Cent
September	1.8
October	7
November	11.9
December	17.9
January	20.2
February	15.2
March	13.2
April	7.9
May	3.9
June	1.0
Total =	100.0

perature control by means of individual room controls (explained in Example 1) which allow the heaters to operate only when heat is actually needed.

Using 6,300 degree-days, the 10 per cent figure just explained, and Table 84, the following analysis can be made, keeping in mind it is an *estimate*. However, the estimating method is surprisingly accurate.

Total estimated heating load is 16 kilowatts, Table 83. Ten per cent of 6,300 degree-days is 630. Multiplying 630 by 16, the kilowatt load, gives 10,080 kilowatt hours as the total season's current consumption. The monthly distribution and costs are shown in Table 85.

Table 85. Estimated Season's Cost of Operation for Electric Heaters, as Shown in Example 2

Months	Per Cent of Total	Kw. Hr.	Cost @ 2c Kw. Hr.
September	1.8	181	$ 3.62
October	7	706	14.12
November	11.9	1,200	24.00
December	17.9	1,804	36.08
January	20.2	2,036	40.72
February	15.2	1,532	30.64
March	13.2	1,331	26.62
April	7.9	796	15.92
May	3.9	393	7.86
June	1	101	2.02
Total	100.0%	10,080	$201.60

PRACTICE PROBLEM

This problem consists of designing an electric heating system for the one-story residence shown in Fig. 124. The heaters are to be of the wall type, automatically controlled. Use the B.t.u. Method to calculate heat losses for all rooms including the laundry.

Note: Fig. 124 shows radiators and a boiler. In this problem these items can be ignored. Also assume frame walls instead of brick.

The problem should be solved by calculating or selecting the following items:

(a) Heat loss in B.t.u. for all rooms.

(b) Wall Heaters for every room.

When items *a* and *b* have been solved, the following item should be worked.

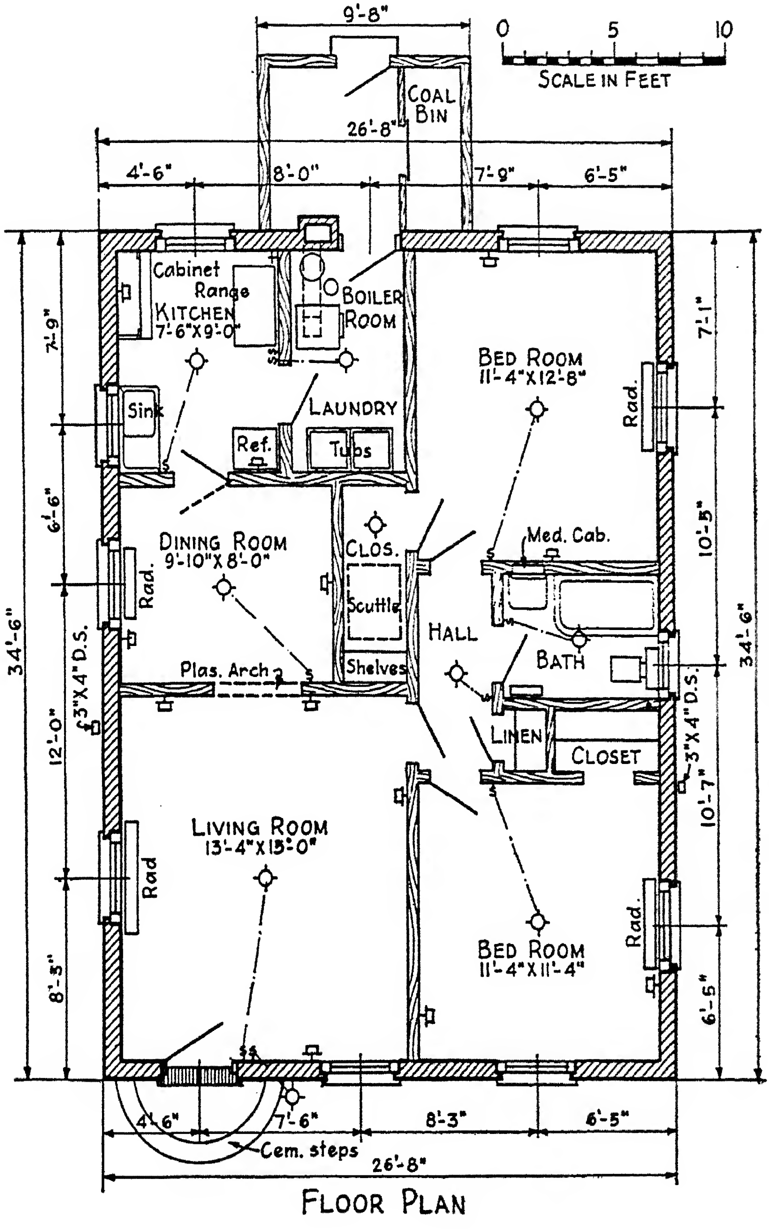


Fig. 124. Floor Plan

(c) Draw a floor plan (scale $\frac{1}{4}'' = 1'0''$) and show the location of all heaters and thermostats for each room. Indicate size of each heater.

Specifications:

Doors: All outside doors are of wood $1\frac{5}{8}$ inches thick. Assume .90 B.t.u. per hour per degree Fahrenheit per foot of crack. All doors are 3 feet by 7 feet.

Windows: All windows single glass and double-hung. Assume $\frac{1}{16}$-inch crack and $\frac{3}{64}$-inch clearance. No weatherstripping. All windows are $3'0'' \times 4'6''$, except the window over the kitchen sink, which is $3'0'' \times 3'0''$, and the bathroom window, which is $2'0'' \times 2'0''$.

Walls: Western Frame. On the outside, sheathing, paper, and siding. On the inside, $\frac{1}{2}$-inch Insulite plaster backing and plaster. Table 14 gives k value for the insulation.

Floor: $\frac{5}{8}$-inch oak flooring over 1-inch Insulite. Under the Insulite is $\frac{7}{8}$-inch yellow pine rough flooring.

Ceiling: 1-inch Insulite plaster backing and plaster on joists. Tight $\frac{5}{8}$-inch yellow pine floor above.

Note: Assume attic temperature is 5°F. above outside design temperature.

Basement: There is no basement. The house sets on concrete foundations so that the floor joists are 18 inches above the ground. There are vents in the foundation, so this area must be assumed to be the same temperature as outside.

Special Note: Assume door between laundry and coal bin room is an outside door. Do not include coal bin room in calculations nor design a heater for it.

Location: Chicago, Ill.

Inside Temperature: Calculate exact temperatures, assuming a breathing line temperature of 70°F. at 5 feet above floor. Review illustrative Example in Vol. II, Chapter VI.

CHAPTER X

HUMIDIFICATION

It is a well-known fact that during a heating season the air in residences, offices, or other inclosures for human occupancy, becomes dry and uncomfortable, unless some provision is made to supply more moisture (water) to the air other than it would ordinarily possess. A brief analysis of an ordinary condition, such as might happen in any locality where there are low winter temperatures (assuming that ordinary heating apparatus and not air conditioning is used), will show why dryness and resultant discomfort occur.

Cause of Dry Air. During the time a residence is being heated, for example, outside air is being constantly received on the inside because of infiltration. Under average conditions one pound of this outside air at a temperature of 0°F. contains approximately 5.60 grains of moisture. This same air, when raised to 70°F. temperature in the rooms, has a relative humidity of approximately 10 per cent. (See Psychrometric Chart at the back of the book.) Most authorities agree that a residence requires a relative humidity of about 45 per cent when the temperature is 70°F. This relative humidity is deemed most healthful where human occupancy is concerned. The difference in moisture content of 70°F. air at 45 per cent relative humidity and 0°F. air at 10 per cent relative humidity is the difference between the grains of moisture per pound for each case. Thus at 70°F. and 45 per cent relative humidity, air contains 49.73 grains of moisture per pound. Air at 0°F. (saturated) contains 5.60 grains of moisture per pound. Then $49.73 - 5.60 = 44.13$ grains difference.

The above illustration serves to show why the air becomes dry in a residence or in any heated enclosure. It might be further explained that because of infiltration the entire volume of air in an inclosure is constantly changing and the rate of change is at least once every hour. As new air comes in, the old air goes out. Thus with the constant inflow of cold air at approximately 10 per cent relative humidity, the dry condition is quickly brought about.

Results of Dry Air. Air has a great affinity for moisture. It is constantly taking up enough moisture to cause a saturated condition. Thus if air in an inclosure becomes dry (low relative humidity), it naturally takes up what moisture there is in the room. This is a natural law of nature that never varies. It follows, therefore, that any available moisture will be taken up by dry air.

In a residence the woodwork, plaster, furniture, plant life, etc. contain some moisture. Dry air attacks these items and takes up much of their moisture. The result is that the woodwork warps, furniture comes apart at glued joints, plaster cracks, and plant life dies.

The occupants of a room containing dry air also suffer. A human body perspires the same in winter as in summer, although the winter perspiring is not noticeable. The perspiring, however, goes on at a constant rate. This perspiration is in the form of moisture which evaporates rapidly and is taken up by the air. Before evaporation can take place, however, heat is required (heat of vaporization). Such heat comes from the skin. A constant drain of skin heat causes an occupant to feel cold even in a room where the temperature is 70° to 85°F.

Another way in which occupants suffer from dry air is through their breathing channels. Air going in the nose during ordinary breathing extracts moisture from the delicate nasal membrane and causes it to become slightly feverish. As a result germs form, which produce the common cold and other bothersome ailments.

A great many more disadvantages of dry air could be enumerated but those given amply illustrate the conditions.

Amount of Moisture Required. The following two methods are commonly used for calculating the amount of moisture in terms of gallons of water, needed for residential humidification. Either method can be accurately employed. The second method however assumes an inside temperature of 69°F. which makes it impossible to use for any other inside temperature. Also, the second method assumes 1½ air changes per hour, whereas the first method may be used for any number of changes.

Note: It is not within the scope of this book to explain insulation. The reader, however, should make sure he understands how to ascertain if the walls of a residence, for example, are well enough insulated so that condensation will not occur during periods of severe weather. Double glazing for windows should be studied where relative humidities of 40 to 45 per cent are planned.

First Method of Moisture Calculation. Assume a residence as having about 25,000 cubic feet in volume with an inside temperature of 70°F. The relative humidity must be 45 per cent when the outdoor temperative is 0°F.—saturated. To calculate the amount of moisture that must be added to the air, the following method is used.

From the Psychrometric Chart, found at the back of book, it can be seen that air at 70°F. and 45 per cent relative humidity contains 49.73 grains of water per pound and that air at 0°F. and saturated contains 5.60 grains per pound. Thus the amount of water to be added is 49.73—5.60=44.13 grains per pound.

A residence of good construction is assumed to have one change of air per hour due to infiltration.

A volume of 25,000 cubic feet of air at one change per hour will weigh roughly 75 pounds per thousand cubic feet (see Psychrometric Chart), or a total of 25×75=1,875 pounds. This is the weight of the air in the residence. Now, if one pound of air should contain 44.13 grains of water, 1,875 pounds should contain 1,875×44.13=82,744 grains. This is the total grains of water that should be added to the air in the residence per hour. There are 7,000 grains of water to a pound. So, 82,744÷7,000=11.82 pounds of water per hour needed in the residence to maintain a relative humidity of 45 per cent. One gallon is equal to 8.33 pounds where water is concerned. Then 11.82÷8.33 =1.42 gallons. This is the number of gallons of water needed to properly humidify the residence for one hour. For one day this would be multiplied by 24 hours or 1.42×24=34 gallons. Thus 34 gallons of water would be required per day to keep the residence properly humidified.

Having found the above figures, based on good construction, they can be increased by whatever percentage is deemed advisable. This depends, however, entirely on the judgment of the designing engineer. For a poorly constructed house, allowance must be made for greater infiltration.

It should be remembered that for conditions where the outside temperature is higher than 0°F. the entrained water will be greater and not as much water per hour would be needed to properly humidify the residence. However, a humidifier should be designed to be capable of handling extreme conditions. For example, the lowest temperature ever recorded in Chicago was -23°F., but for design purposes a temperature of 0°F. could safely be used, because it is not likely that the low temperature record would be duplicated very often—perhaps a day or two at a time.

* *Second Method of Moisture Calculation.* Fig. 125 shows a chart from which can be determined the number of gallons of water per hour needed for humidification of 10,000 cubic feet of air. The inside temperature is 69°F. and the outdoor temperature anywhere between -10° and 70°F. The curves for 35, 40, and 45 per cent relative humidity are based on 1½ air changes per hour.

For example, Fig. 125 can be used to determine how to humidify a residence of 15,000 cubic feet of heated space so that it will have a humidity of 45 per cent inside, with the inside temperature 69°F. and outside temperature 0°F.

* "Humidity Requirements for Residences," by Prof. A. R. Kratz, Transactions A.S.H.V.E. 1923.

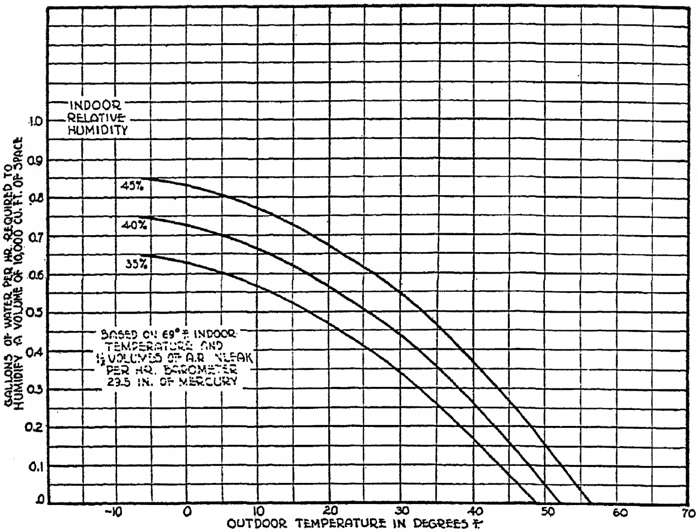


Fig. 125. Requirement in Terms of Gallons of Water for Residence Humidification

Locate the outside temperature 0°F. along the bottom of the chart. Follow a vertical line upward from 0°F. until the line intersects the 45 per cent curve. From the point of intersection, follow a horizontal line to the left-hand side of the chart and read 0.83. Then, because the volume required is half again as much as 10,000, the largest number recorded on the chart, multiply 0.83 by 1.5 which equals 1.25. Therefore the residence will require 1.25 gallons of water per hour or 1.25×24=30.00 gallons for one day.

Note: It should be kept in mind that the second method assumed 1½ complete changes of air per hour whereas the first method assumed only 1 change per hour.

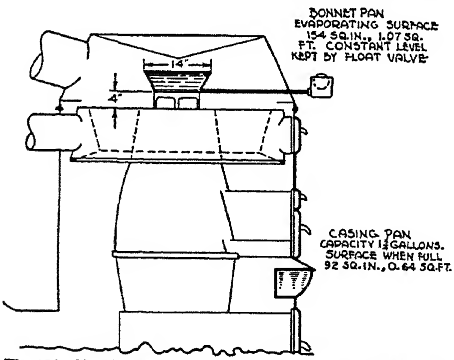


Fig. 126. Sketch of a Gravity Hot-Air Furnace Showing Casing and Bonnet Water Pans

Residential Humidification. If winter or the all year round air-conditioning apparatus is used for heating, the humidity problem is amply taken care of. The following discussion, as noted at the beginning of this chapter, applies only where ordinary heating systems are employed.

Gravity Furnace Method. There are various ways in which humidity apparatus can be used in connection with a hot air furnace. Three of the best known methods will be described here.

The oldest method consists of placing a pan just inside the outer casing and not far above the base of the furnace. See Fig. 126.

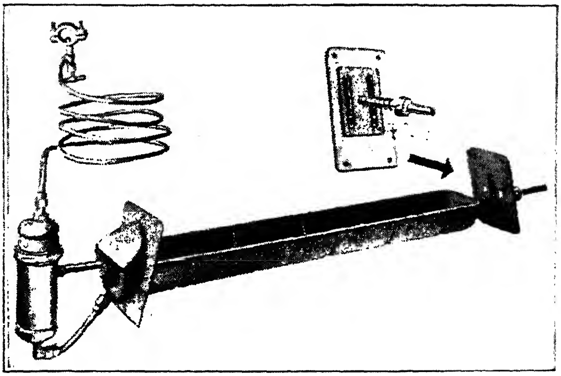


Fig. 127. Maid-O'Mist Humidifier for Installation in Hood or Bonnet of Furnace
Courtesy of Maid-O'-Mist, Inc., Chicago, Illinois

The circulating air in passing over and around the water takes up some of it and circulates it through the residence. The amount of water taken up by the air depends upon the area of water surface in the pan and the velocity of the air.

Another method is called the bonnet pan method. The pan is placed in the bonnet of the furnace, as shown in Fig. 126. The pan should be directly over the combustion chamber. The water is supplied to the pan automatically, as needed, by a float valve. Tests have shown that the bonnet pan method is superior to the casing pan method in regard to the amount of water evaporated.

A third method employs an apparatus, as shown in Fig. 127, which is also placed in the bonnet of the furnace.

The humidifier pan is made of one piece of "Naval Bronze,"

which metal, the Navy has found to be the least corrosive of any metal practical for a unit containing water. The unit is provided with adjustment plates in the rear, which permit the user to lift the pan to the first notch to cut the capacity of the pan to ¾, and to the top notch to cut the capacity of the pan to ½, to regulate humidity in order to suit specific conditions.

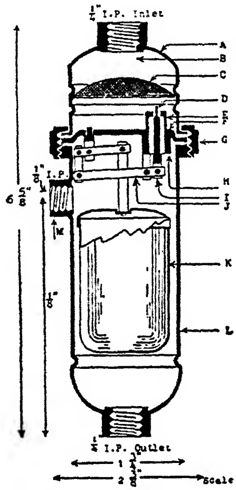


Fig. 128. Water-Boy Safety Water Feeder
Courtesy of Maid-O'-Mist, Inc., Chicago, Illinois

An automatic water feeder, Fig. 128, is installed in front of the pan, away from the heat of the furnace, which further wards off any liming or corrosion. All working parts are made of non-ferrous materials, and the unit is provided with a 100-mesh monel screen to catch any foreign matter in the water. A nickel silver valve and valve seat are used in the mechanism which will stand hard usage. The installation is not complicated, and only necessitates cutting two holes in the hood or bonnet of the furnace, and drilling a small hole in a water line which is close to the furnace. Other than this it is only necessary to assemble the unit.

One pan is usually sufficient for the average furnace, but it is recommended that two pans be connected to one feeder where a very low bonnet temperature is maintained, such as is often the case on a

gas-fired job. Either pan may be adjusted to cut down the capacity, independent of the other.

Residences, to be efficiently humidified must be reasonably tight; that is, the outside doors, window sashes and frames must be close fitting and there must be little leakage of outside air at other points.

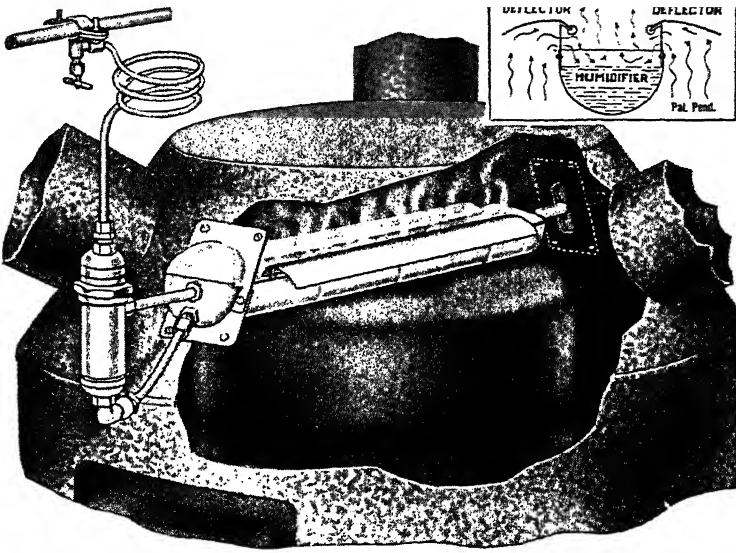


Fig. 129. Maid-O'-Mist Zephyr Automatic Humidifier
Courtesy of Maid-O'-Mist, Inc., Chicago, Illinois

Failure to carry out these specifications would cause a residence to exceed the capacity of the humidifier for even 30 per cent humidity, not to mention 40 per cent.

Outside walls must be of good construction. Windows should be double glazed. If the walls are poorly insulated and the windows of single glaze, there may be excessive condensation which, besides being unsightly, may cause much damage to decorations.

Mechanical Furnace Method. For mechanical furnaces having a greater and more positive air supply, humidification can be accomplished in a more exact manner. Fig. 129 shows one type of an automatic humidifier designed for mechanical furnaces or forced circulation warm air furnaces. The pan is 4x36 inches in length and stamped from sheet bronze. The heating area exceeds 350

square inches. The pan is equipped with patented wings to plane the air over the water surface. This action is shown in the insert in Fig. 129. The hood is adjustable and fits the slope of any furnace.

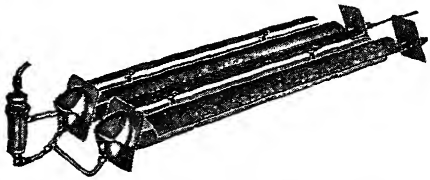


Fig. 130. Twin Zephyr Automatic Humidifier
Courtesy of Maid-O'-Mist, Inc., Chicago, Illinois

A self-locking overflow plate permits the raising and lowering of one end of the pan to reduce evaporating capacity in extreme cases. The Zephyr is supplied with a feeder, like the one shown in Fig. 128.

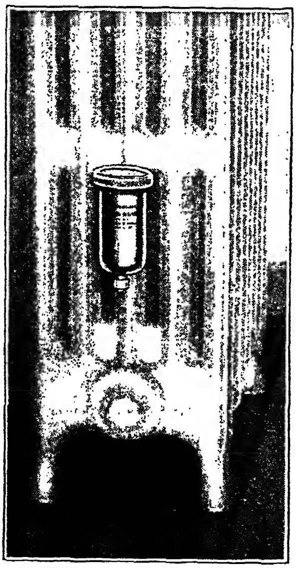


Fig. 131. Auto-Vent Humidifier for Use with Steam Radiators
Courtesy of Maid-O'-Mist, Inc., Chicago, Illinois

The twin automatic humidifier, Fig. 130, has been designed for very large mechanical furnaces or where a high percentage of humidity is desired. One feeder supplies both pans, but each pan can

be lowered or raised as in Fig. 129. The wings, area, etc., per pan are the same as Fig. 129.

Humidification in a Steam Heating System. Fig. 131 shows a Maid-O'-Mist Auto-Vent used in place of an ordinary vent on a steam radiator. The auto-vent supplies approximately one pint of water vapor to the air per hour when the steam pressure is one

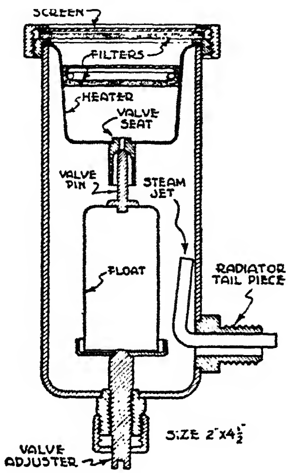


Fig. 132. Sectional View of Auto-Vent Humidifier
Courtesy of Maid-O'-Mist Inc., Chicago, Illinois

pound. Fig. 132 shows a cross-sectional view of the auto-vent in which the parts have been named. This humidifier is typical of the principle of supplying humidity by means of the steam in the system. The amount of humidity can be controlled by the adjuster, Fig. 132.

Mechanical Humidification of Residences. Fig. 133 shows a type of humidifier of a mechanical nature which filters and circulates the air. It can be used in connection with furnace, hot water, steam, electric, etc., types of heating systems. The humidifier consists of a metal casting, Fig. 134, in which is located a small coil heated by steam, hot water, or vapor, depending on the system installed. For residences having a hot air furnace system, the mechanical humidifier is more expensive to use, as the coil is supplied with heat from some other source than direct from the heating system.

The steam supply, for example, maintains the coil at a high temperature. Water is sprayed on to the coil by means of a spray

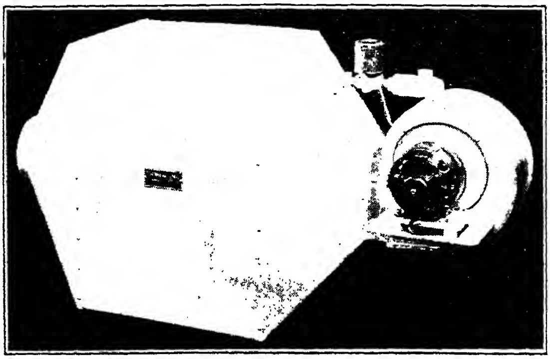


Fig. 133. Automatic Humidifier
Courtesy of Thermal Units Manufacturing Company, Chicago, Illinois

which is controlled by a solenoid valve. The solenoid valve is controlled by a humidostat located in the living quarters of the resi-

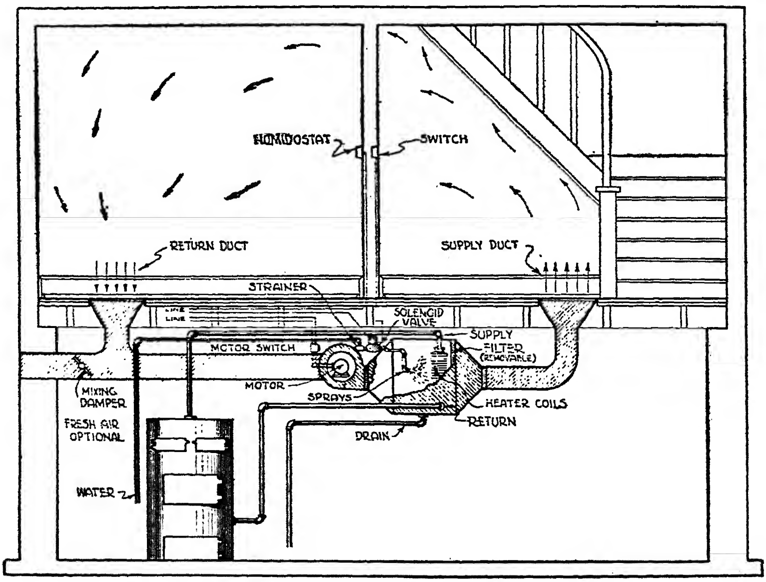


Fig. 134. Typical Installation of Mechanical Humidifier in Residence

dence. As the spray water comes in contact with the hot coil it turns to steam. This steam is then absorbed by the air and pro-

pelled through the supply duct by a motor-driven fan. The moistened air circulates through the residence and comes back to the humidifier by way of the return duct where it is strained and cleaned before starting the cycle over again. The humidostat operates to keep the relative humidity at an exact point and turns the humidifier fan on and off at intervals as required. Fresh air can be circulated

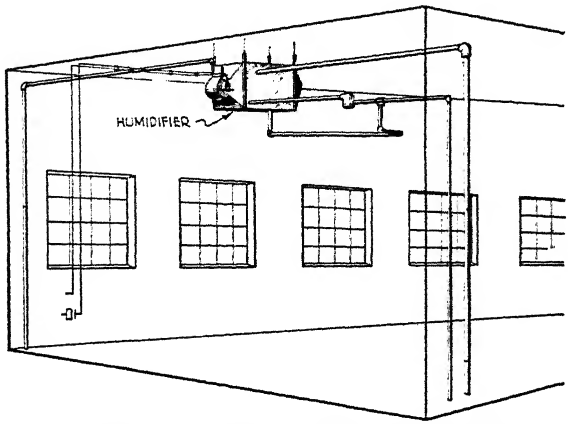
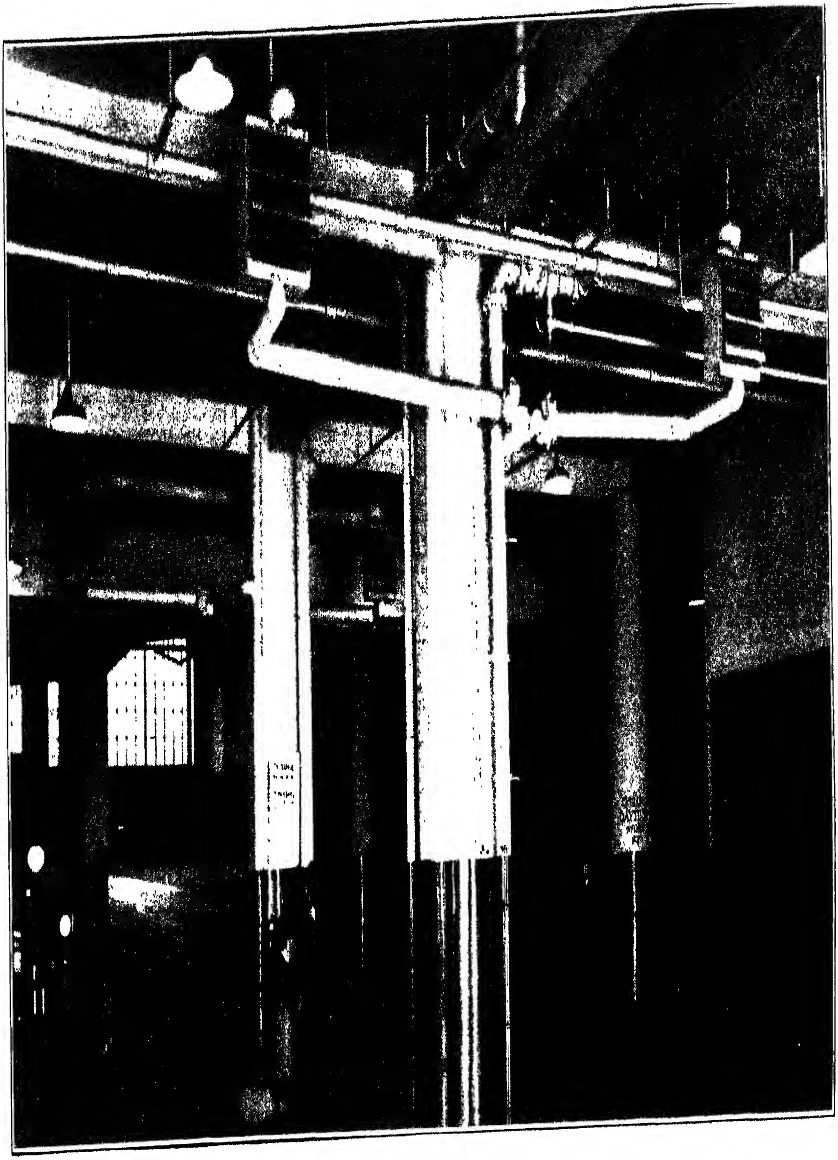


Fig. 135. Typical Installation of Mechanical Humidifier for Industrial Building

constantly if desired, as shown in Fig. 134, by the optional fresh air duct and the mixing damper. The humidifier can be used without ducts, but in this case must be installed directly in the area to be humidified.

Mechanical Humidification for Industrial Buildings. For large offices, factory areas, and other industrial areas, mechanical humidification can be accomplished by using the same type of humidifier as shown in Fig. 133. For industrial applications, the humidifier is generally suspended from the ceiling as shown in Fig. 135. From such a position it functions exactly as for an installation where ducts are employed. The operation of the humidifier is controlled by automatic regulators.

Selection of Humidifier Size. The manufacturers of mechanical humidifiers publish catalogues giving the various sizes together with evaporating capacities per hour.



UNIT HEATER INSULATION

Courtesy of McMay, Incorporated

CHAPTER XI

UNIT HEATERS

One of the interesting mechanical developments of the age is the adoption of production methods in manufacturing. One result of such methods of course is the turning out of an enormous volume of the product at low cost.

Called upon to design the heating of an industrial plant, the engineer is quite likely to employ unit type heaters, which are usually assembled on a production line and delivered ready to operate as soon as power, controls, and piping are connected. In many cases units are more efficient than a central heat and air distributing system because heat can be transported with less loss by small insulated pipes than by large ducts, and because of the economy with which electrical energy may be distributed.

For instance, less power is required to deliver 15,000 cubic feet per minute of air to 10 different rooms or zones with 10 unit heaters at, say, 1,500 cubic feet per minute each, than is required to deliver 15,000 cubic feet per minute to the same rooms or zones through ducts. The 10 units may consume something like 2.5 horsepower while the single unit may require as much as 4 horsepower. When the filtering and warming and the controlling of temperature of air introduced for ventilating purposes from out-of-doors is considered, of course the complication and cost involved in providing and controlling a number of separate units with their various intake openings, air filters, tempering heaters and the piping, valves and electric wiring, may offset the saving that otherwise might be made. Thus, a study of the individual case is required in order that the proper system may be chosen.

Unit heaters are manufactured in many varieties but their basic principles are similar in that forced air is heated by passing through coils which are supplied with hot water or steam, as a heating medium. In rare instances gas is used to heat water for the coils, but in most cases the heating medium is supplied from a heating system boiler of usual design. Unit heaters are most generally used

in industrial buildings although offices can be heated by them successfully.

Kinds of Unit Heaters. The most common type of unit heater is shown in Fig. 136. This type is generally hung from the ceiling by steel pipe hangers or straps. In this position the heater draws

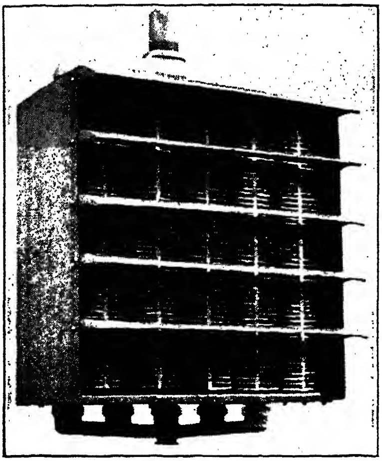


Fig. 136. Typical Unit Heater of the Type to Be Hung from Ceiling
Courtesy of Thermal Units Manufacturing Company, Chicago, Illinois

air from about the same level and discharges it downward into the space occupied by offices or manufacturing processes in industrial buildings.

Another type of unit heater is shown in Fig. 137. This type is mounted on legs and in operation draws cold air from the floor, heats it, and discharges it above the areas occupied by workers or manufacturing processes.

Unit heaters have many advantages including low cost, mobility, and small capacities. Standard designs, whereby no individual designing for given installations is necessary, together with large production, have brought about the advantages of low cost. Added to this is the fact that no ducts need be installed. The heaters can be readily moved when necessary to allow for alterations in offices, plants, etc., and to take care of shifting heat needs. The small capacities of these heaters enable them to be used readily by small

offices or shops and thus avoid the necessity of installing central systems that would take up a large amount of space and not prove economical.

Technical Data for Unit Heaters. Unit heaters are a combination of heating coils and a motor-driven fan enclosed within a metal

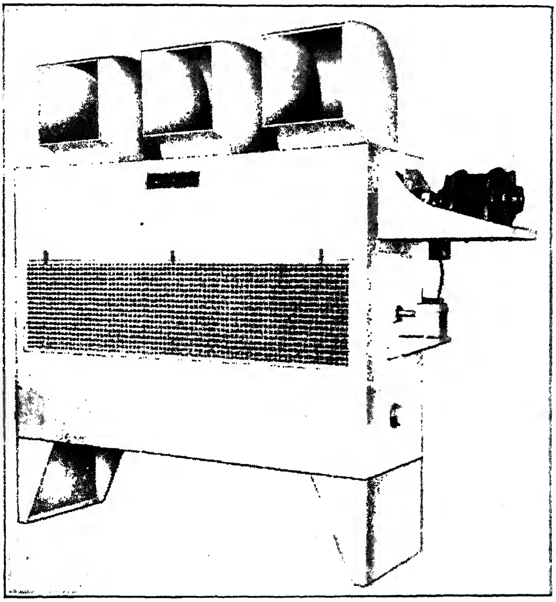
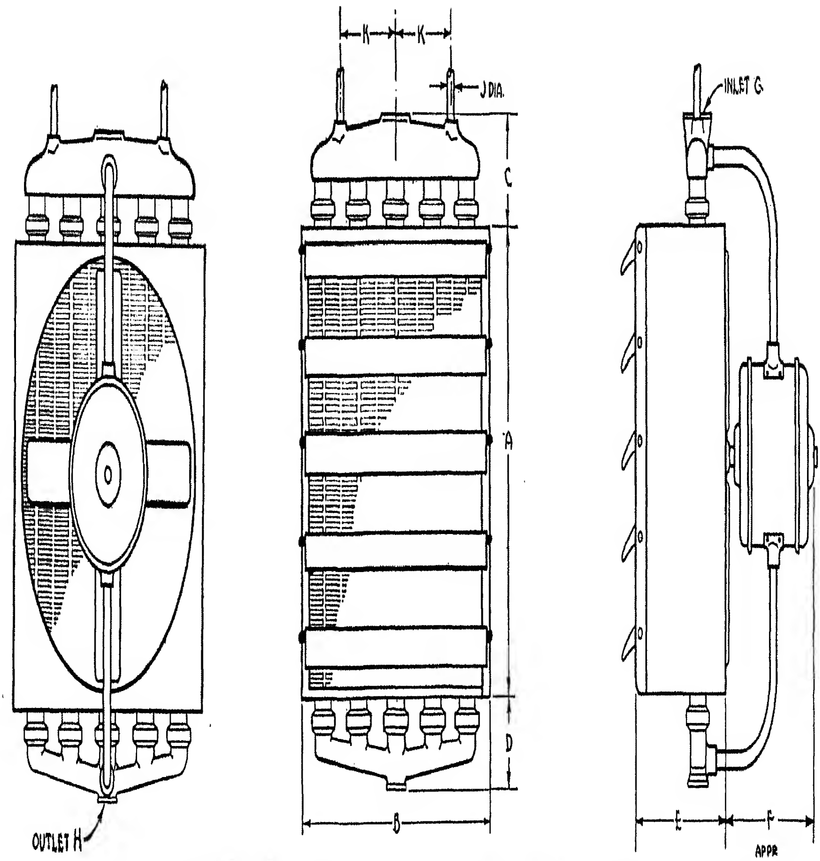


Fig. 137. Floor Type Unit Heater
Courtesy of Buffalo Forge Company

casing and having louvers for controlling direction of discharged air. Some unit heaters, such as shown in Fig. 137, have outlets to give definite direction to the discharged air thus increasing their general efficiency.

Fig. 138 shows a front, side, and rear view of a typical unit heater together with 5 typical sizes and dimensions. Table 86 shows typical capacities for the 5 sizes given in Fig. 138. For example. unit number 20 is 21⅞₆ inches wide, 10⁹⁄₁₆ plus 10 inches in over-all thickness, and has an inlet of 2 inches. At 1,150 r.p.m. and a nominal horsepower of ⅙, it will deliver 142,900 B.t.u. per hour under 5 pounds of steam and 3,120 cubic feet per minute of air. These dimensions and ratings are used in planning space and selecting the size of heater required for a given case.



Unit No.	A	B	C	D	E	F	G	H	J	K
12	13 5/8	13 1/4	4 1/4	3 1/2	10 5/16	10	1 1/4	3/4	1/2	3
16	17 5/8	17 1/4	4 7/8	3 3/4	10 3/8	10	2	3/4	1/2	4
20	21 9/16	21 7/16	4 3/8	3 3/4	10 9/16	10	2	1	5/8	6
24	25 5/8	25 1/4	4 7/8	4	11 3/8	10	2 1/2	1 1/4	5/8	6
30	31 3/4	29 3/8	4 15/16	3 15/16	12 7/8	10 1/2	2 1/2	1 1/4	3/4	8

Fig. 138. Front, Side, and Rear Views of a Typical Unit Heater and Dimensions for Five Common Sizes

Courtesy of Thermal Units Manufacturing Company, Chicago, Illinois

Table 86. Capacity Table

Model	R.P.M.	Nom. H.P.	C.F.M. Air Measured at 70° F.	Outlet Air Vel. Feet. Per Min. At Outlet Temp.	Ent. Air Temp.	Hot Water Aver. Temp. in Coil 170° F. B.T.U./Hr.	F.T.	Atmosph. Pressure (Vapor Systems) B.T.U./Hr.	F.T.	Steam Pressure at Heater 1# B.T.U./Hr.	F.T.	2# B.T.U./Hr.	F.T.	5# B.T.U./Hr.	F.T.	10# B.T.U./Hr.	F.T.	30# B.T.U./Hr.	F.T.
12	1750	1/10	1120	1165	50	30600	75	47000	89	48000	90	49000	91	51400	93	55000	96	65000	104
			1101	1159	60	28200	84	43300	97	44200	98	45100	99	47600	100	51000	103	60900	112
			1079	1152	70	25800	92	39700	104	40500	105	41300	106	43900	108	47300	111	56900	119
	1150	1/20	935	976	50	26700	77	41000	92	41900	93	42300	94	45800	96	49200	99	57900	108
			917	970	60	24600	85	37800	99	38600	100	39400	101	42400	103	45600	107	54300	115
			899	964	70	22500	94	34600	107	35300	108	36000	109	39100	111	42300	114	50800	123
	850	1/20	572	603	50	19600	82	30200	99	30800	100	31400	101	33000	104	35500	108	41800	118
			561	599	60	17900	90	27700	106	28300	107	28850	108	30500	111	32900	115	39200	125
			550	595	70	16600	98	25500	113	26000	114	26500	115	28200	118	30500	122	36600	132
16	1150	1/6	1997	1165	50	54400	75	83500	89	85200	90	87000	91	91600	93	98000	96	115100	104
			1958	1159	60	50600	84	77800	97	79400	98	81000	99	83600	100	[illegible]	103	[illegible]	112
			1920	1152	70	45700	92	70200	104	71700	105	73100	106	77900	108	[illegible]	111	100400	119
	850	1/10	1664	976	50	48700	77	74800	92	76400	93	77900	94	81700	96	87000	99	103000	108
			1632	970	60	44400	85	68300	99	69600	100	71000	101	74800	103	81800	107	95700	115
			1600	964	70	41400	94	63600	107	64900	108	66200	109	79000	111	75200	114	90500	123
	550	Cond. Type	1018	603	50	34700	82	53300	99	54400	100	55500	101	58500	104	63100	108	74000	118
			998	599	60	32000	90	49100	106	50100	107	51100	108	54400	111	57600	115	69300	125
			980	595	70	29400	98	45100	113	46000	114	47000	115	50200	118	54400	122	64900	132
20	1150	1/6	3120	1165	50	85000	75	130700	89	133400	90	136100	91	142900	93	152800	96	180800	104
			3060	1159	60	78400	84	120300	97	122800	98	125300	99	132200	100	[illegible]	103	[illegible]	112
			3000	1152	70	71900	92	110400	104	112700	105	115000	106	122100	108	[illegible]	111	[illegible]	119
	850	1/10	2600	976	50	75700	77	116100	92	115700	93	121100	94	127300	96	136700	99	161100	108
			2550	970	60	69700	85	107100	99	109300	100	111500	101	117800	103	126900	107	150900	115
			2500	964	70	64000	94	98300	107	100300	108	102300	109	108600	111	117600	114	141100	123
	550	Cond. Type	1590	603	50	53600	82	82500	99	84100	100	85800	101	91800	104	98600	108	110200	118
			1560	599	60	50200	90	77200	106	78800	107	80400	108	84900	111	91500	115	105900	125
			1530	595	70	46100	98	71000	113	72400	114	73800	115	78300	118	84800	122	101800	132
24	1150	1/4	4493	1165	50	122200	75	187900	89	191800	90	195600	91	206200	93	220600	96	255900	104
			4406	1159	60	114000	84	175000	97	178600	98	182200	99	188000	100	202100	103	244300	112
			4320	1152	70	102800	92	158200	104	161400	105	164900	106	175200	108	189100	111	225900	119
	850	1/6	3744	976	50	109000	77	168400	92	171800	93	174200	94	183800	96	195800	99	231800	108
			3672	970	60	100000	85	153500	99	156600	100	159700	101	168400	103	184100	107	215400	115
			3600	964	70	93100	94	143100	107	146000	108	148900	109	157600	111	169100	114	203700	123
	550	Cond. Type	2290	603	50	77900	82	119700	99	122200	100	124600	101	132000	104	141800	108	156300	118
			2246	599	60	71800	90	110300	106	112600	107	114800	108	122200	111	131800	115	155700	125
			2203	595	70	66000	98	101300	113	103400	114	105500	115	112800	118	122300	122	145800	132
30	1150	1/2	6552	1165	50	178000	75	274000	89	279700	90	285300	91	300700	93	321700	96	[illegible]	104
			6426	1159	60	166000	84	255200	97	260400	98	[illegible]	99	274100	100	294700	103	356400	112
			6300	1152	70	150000	92	230700	104	235400	105	240100	106	255500	108	275700	111	329500	119
	850	1/3	5460	976	50	159800	77	245900	92	[illegible]	93	255600	94	[illegible]	96	285500	99	338000	108
			5355	970	60	145700	85	223800	99	228400	100	233000	101	245600	103	263400	107	314100	115
			5250	964	70	135800	94	208600	107	212900	108	217200	109	229800	111	[illegible]	114	297000	123
	550	Cond. Type	3339	603	50	113300	82	174200	99	177800	100	181300	101	192000	104	206200	108	241800	118
			3276	599	60	104600	90	160500	106	163800	107	167100	108	177700	111	191700	115	226500	125
			3213	595	70	96000	98	147500	113	150500	114	153500	115	164100	118	177800	122	212000	132

The component parts of a heater, as shown in Figs. 136 and 138, are the heating element, motor, fan, and louvers assembled into one unit and enclosed in a sheet metal cabinet which can be painted any desired color.

Fig. 139 shows a cutout view of the heating coil for a heater such as is shown in Fig. 136. This coil may be supplied with hot

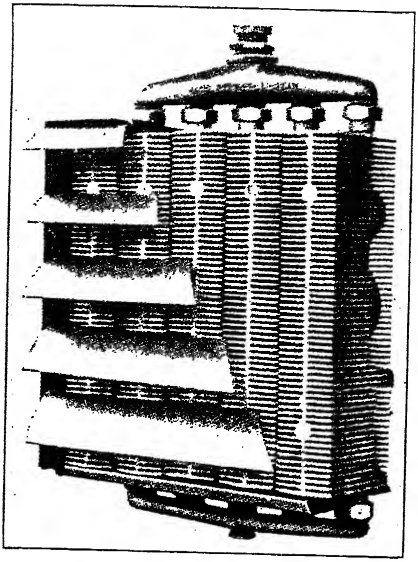


Fig. 139. Cutout View of a Unit Heater Showing Coils
Courtesy of Thermal Units Manufacturing Company, Chicago, Illinois

water, vapor, or steam as a heat medium. Hot water should have a temperature of 170°F., at least, in the coil. Steam can vary between 1 and 250 pounds at the coil, depending on the system used. The operating principle of the coils is the same as for an ordinary hot water or steam radiator, except that the fan makes the process more rapid. The motors and fans vary in size with the complete unit but otherwise are standard types designed for silent operation.

The principle of the airplane wing has been used in designing the louvers. They direct the air current as it leaves the unit, and are spaced and hinged separately to secure various angles of deflection for better control of the heated air stream.

The unit motors can be started and stopped by manual operating switches or controlled by thermostats.

**Air Temperatures.* In the case of suspended unit heaters that use air at some distance above the floor, the temperature variation from floor to ceiling may reach one degree for each foot space (vertical) between the heater and the floor during periods when the heater is put to the greatest use. Thus this allowance should be made in calculating the capacity of suspended heaters. Heaters taking in recirculated air at the floor level should maintain temperature differentials of less than .5°F. per foot of elevation when the maximum capacity of the heater is required.

For calculating heaters with intakes at floor levels, the temperature to be maintained in the room should be considered as the temperature of the air entering the heater. Where outside air is introduced, the temperature of the mixture must be calculated and used as the entering air temperature to the heater. For suspended heaters, a higher entering air temperature should be used than that at which the heat in the room is to be maintained.

Unit heaters save fuel because they circulate air at a lower average temperature than the air circulated by direct radiators; however, the heaters must circulate more air in a given time than would be needed with direct radiators. For this reason heaters with ample air capacity must be carefully selected. Very low temperatures can be had only at the expense of larger heaters and increased power. It is advisable to use a delivery temperature of 70°F. above the room temperature desired.

Boiler Capacity. The capacity of a boiler to use in conjunction with unit heaters must be based on the capacity of the heaters at the lower entering air temperature. In most cases, this is taken as 40°F. For cases where air is taken from the outside, the temperature should be about the same as the extreme low for the locality.

Good practice dictates that it is not a good policy to use only one heater as the load for a boiler. It is best to employ two smaller heaters. This is especially true where automatic control is used, as the sudden changes of load that would occur under such conditions would require too much attention to the boiler.

Connections. For unit heater installations, a one-pipe gravity

* A.S.H.V.E. Transactions, Vol. 39, 1933.

or vapor system does not work out as well as a two-pipe closed gravity return system. Fig. 140 shows typical connections.

Selection of Unit Heaters. The process of selecting unit heaters requires the calculation of heat losses in the same manner as for other heating systems, except that unit heaters have a tendency to reduce the temperature difference between the floor and ceiling. Heaters may be so arranged as to recirculate the air, or to supply and warm the air from the outside. In cases where part of the air is to be taken from the outside, the heat necessary to warm the cold air must be added to the sum of the heat losses.

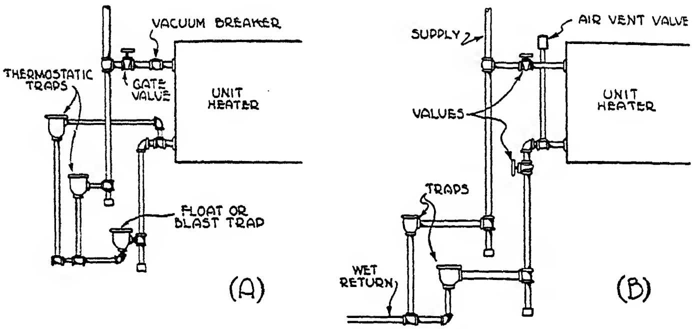


Fig. 140. (A) Unit Heater Connections Where Condensation Is Returned to Vacuum Pump or to an Open Vented Receiver; (B) Unit Heater Connections Where Condensation Is Returned to Boiler through Wet Return

Knowing the heat loss and what heating medium is to be used, the heaters can be selected from manufacturers' catalogue data, (see Table 86).

Control of Unit Heaters. With thermally circulating steam or hot water heating, it has been customary, when basement rooms must have heat, to install ceiling type radiators so that they may be placed above the mains. Such radiators are inefficient because of stratification of the warm air overhead, and are unsightly. The radiant heat from them is never pleasant as it beats down from above. Small electric-driven unit heaters solve this difficulty admirably, since they can be placed above the steam or water mains, yet will deliver the warm air downward and alleviate the stratification. A typical control arrangement for such a heater is shown in Fig. 141.

The fan motor may be controlled at full voltage through the

relay or if the motor is of small size the line voltage may be used throughout. The room thermostat may control or the snap switch may bring about operation of the fan as for summer cooling. In winter, should the thermostat demand heat, the motor valve on the

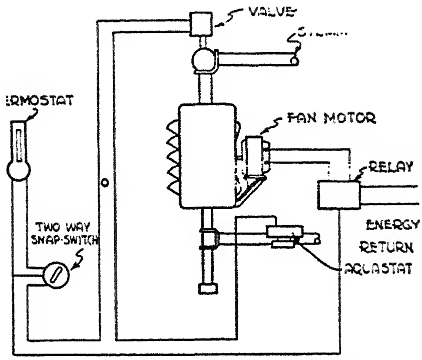


Fig. 141. Control Arrangement for a Recirculating Type Unit Heater

steam supply will be opened, but the fan will not operate (lest it cause cold drafts) until the return pipe, to which the aquastat switch responds, permits energy to pass to the fan motor.

In foundries, machine shops, and heavy manufacturing plants where crane-ways must be kept clear of ducts and other obstructions, recirculating unit heaters are a useful recourse, since they can be placed above the cranes and can force the air downward so as to warm the working spaces. Or, they may be placed under the

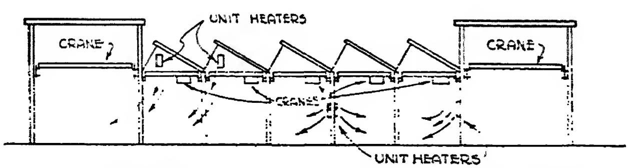


Fig. 142. Cross-Sectional View of a Factory Illustrating Two Methods of Unit Heater Application

crane tracks with horizontal diffusion, where they will not interfere with the free travel of the material in course of manufacture.

At the left of Fig. 142, the units are above the cranes and deliver the air toward the floor at high velocity. Locating the supply and return piping for such an installation may be difficult because

of the crane clearances. At the right, the unit heaters are under the crane-ways. The supply and return piping may be placed directly under the crane girders but if the heating medium is steam, the return piping may have to be (most objectionably) placed under the floor.

Location of Unit Heaters. There are no fixed rules for height above the floor for locating overhead unit heaters, this being a

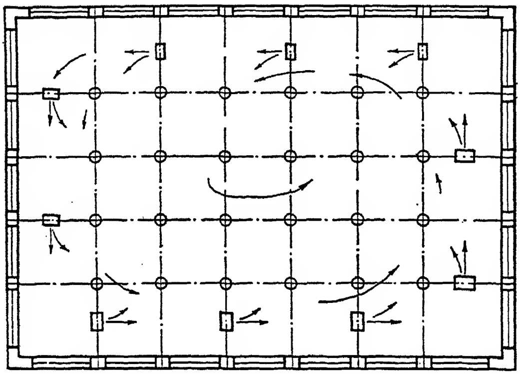


Fig. 143. Typical Location of Small Over-Head Type Unit Heaters for One Large Room

function of the permissible air delivery velocity and this again being reflected by the permissible noise level and by the type of fan employed. Where propeller type fans are used, the units ordinarily give best results if not more than 10 feet above the floor, and if located rather near the outer walls or windows. The machines preferably should pass the air along, each to the other, rather than to oppose each other. In factory service one of these small units every 40 feet around the edges of the room may give excellent results (Fig. 143). The arrangement is such that each heater relays the air toward its neighbor so as gradually to bring about a rotary movement of the room air.

Where centrifugal fans, with their higher outlet velocity and more easily controllable diffusion are employed, the distance between units often is increased to 80 or even to 100 feet depending of course on the refinement of air diffusion which must be maintained. With these units it is not unusual to follow the relayed air circulation shown in Fig. 143.

TYPICAL QUESTIONS AND ANSWERS

A plant maintenance engineer or a reader might ask the following typical questions. Answers are given with the questions.

How high above the floor can units be safely placed?

There are many successful installations with the units placed as much as 40 feet above the floor. The upper limit has never been reached.

What speed of air flow is best, and why?

Speed of air flow is governed by type of occupancy, height of unit above floor, quality of building construction, etc. The air must not strike people at so high a speed as to create discomfort, but some people with their feet on the floor of a room may not be uncomfortable due to air blast from a unit heater 20 feet overhead which has a properly diffused outlet velocity of 2,000 feet per minute. Other people in the same room, however, may be uncomfortable and may have cold feet when a unit heater is only 6 feet above the floor with an outlet velocity of 300 feet per minute. Factories, garages, gymnasiums and the like may very satisfactorily be served by unit heaters, but if these same rooms were used as offices or living rooms, the same arrangement might not be satisfactory. Therefore the question cannot be answered definitely.

What steam pressure is most economical taking into consideration first cost and future cost of operation?

There is a considerable experience and much literature to indicate that the most economical steam pressure for use in unit heaters is the lowest steam pressure that can be circulated; sub-atmospheric during mild weather and never more than 2 pounds above atmospheric pressure. This permits of sub-atmospheric vaporization in the boiler, with the greatest possible temperature difference between the water and the products of combustion.

With sub-atmospheric steam circulation the piping usually is smaller than when higher pressure is used, reducing the investment. High pressure steam, say 50 pounds or so, may be employed in unit heaters, and the piping may be very small under such circumstances, but there is often difficulty in removing the air from the piping and the convectors, and trouble with corrosion due to excess oxygen in the presence of moisture and high temperature.

*ILLUSTRATIVE EXAMPLE

For this example, the reader is referred to Fig. 107 (Factory Building) in Vol. II, Chapter VI. All data relative to this figure, including topographical location, temperatures, structural details, etc., are the same as given in the Illustrative Example for Fig. 107 in Vol. II. It is required to apply unit heating to this building, select heaters, determine their locations, the number of heaters, design piping, indicate control, and design the size of the steam boiler required.

Solution. *Heat Loss.* Ordinarily the first step in the solution of such an example would be the calculation of heat losses. However this has already been done in Vol. II, Chapter VI devoted to the calculation of "Heating Loads." The hourly heat loss is shown as 517,442 B.t.u.

Selection of Heaters. For this case No. 30 unit heaters (see Fig. 139 and Table 86) could be used because two such heaters, with ½ horsepower

* Courtesy of Thermal Units Mfg. Co., Chicago.

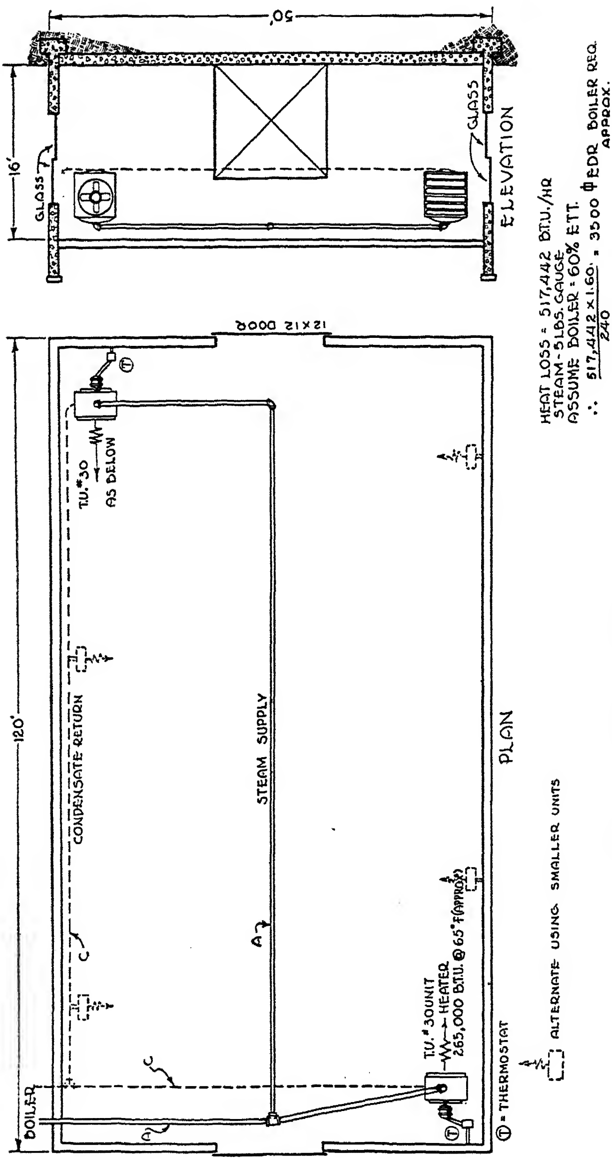


Fig. 144. Arrangement of Heaters and Piping in a Factory Building

motors, each handle approximately 6,400 cubic feet per minute at sufficient velocity to adequately cover the 120-foot length of the building (see Fig. 144). The 2 units delivering 12,800 cubic feet per minute times 60, would give a total of 768,000 cubic feet per hour. This amount, divided by the cubical content of the building, i.e., 96,000 cubic feet, would give about eight re-circulated air changes per hour, or one about every six minutes. This assures evenness of temperature throughout the heated area.

The size of the units was determined so as to use as few as possible, thus making for lower cost of initial equipment and lower installation costs. This is the reason for selecting 2 No. 30 units with a capacity of approximately 265,000 B.t.u. per hour at 65°F. entering air.

The location of the units (see Fig. 144) is such that the air will sweep along the two exposed sides having the glass area, where the greatest heat loss occurs. The units will indirectly throw a stream of heated air to the 12x12-foot doors and thus tend to temper the cold air entering when the doors are opened at frequent intervals.

Design of Piping. Fig. 145 is a diagram and installation chart. The pipe size chart shows that the pipes *A, B, C,* and *D,* as illustrated in Fig. 145, should be 2½, 1¼, 1½, and ¾ inches in diameter, respectively. The runs of supply and return pipes are shown graphically in Fig. 144. All other piping information including valves, scale pockets, traps, etc., are also shown in Fig. 145.

Design of Boiler. Assuming that the average boiler is about 60 per cent efficient, we can multiply 517,442 B.t.u. (heat loss) by 1.6 and divide the quotient by 240 B.t.u. which gives a boiler of about 3,500 square feet equivalent in direct radiation as being required.

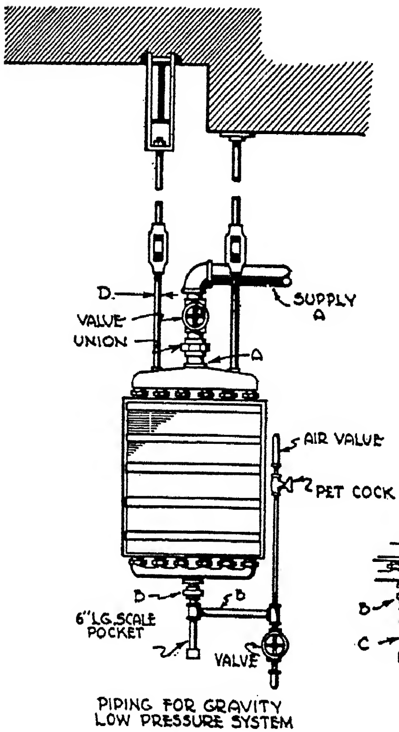
Note: Boilers are explained more in detail in Vol. V.

Design of Controls. To control this installation, use two thermostats as shown in Fig. 144. Care must be taken to see that the thermostats are mounted in the path of the *return* air and not directly in contact with any cold walls. (See Chapter XII on "Automatic Controls.") Electrical connections are shown in Fig. 145. The thermostats should be connected to the motor and fan circuit to stop and start the motors according to the necessary heat required.

The location of unit heaters in a building may be varied considerably to suit conditions and to make for less pipe runs and installation costs. However, the general rule is to direct a stream of heated air along the surfaces where the greatest heat leakage occurs, attempting to *wipe* these surfaces. In a case where recommended unit heaters cannot throw the heated air the entire length of the building, four smaller units would probably be required which could be installed as shown by dotted lines in Fig. 144.

EXAMPLE FOR PRACTICE

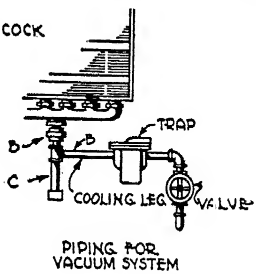
Fig. 146 shows a plan and elevation view of a typical factory building which is to be heated by unit heaters. The windows are each 8′ 0″x6′ 0″. The double door is 1⅝-inch wood and is 8′ 0″x6′ 0″. The walls are common brick and are 12 inches thick. The roof is 1½-inch wood planks covered with a composition (built up) roofing. The floor is 4-inch concrete on ground. The single door is 1⅝-inch wood and is 3′ 0″x6′ 0″. The windows fit well. Assume



THERMAL UNITS MANUFACTURING COMPANY

MODEL Nº	12	16	20	24	30
STEAM PRESSURE	SIZE OF TRAP CONDENS. P. HR. FOR 60° ENT. AIR				
5	65	110	170	250	340
50	100	170	260	375	550
100	120	200	310	450	650
150	130	225	350	500	740
PIPE SIZES					
SUPPLY "A"	$1\frac{1}{4}$	2	2	$2\frac{1}{2}$	$2\frac{1}{2}$
RETURN "B"	$\frac{3}{4}$	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{4}$
SCALE POCKET "C"	1	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{2}$
DIA. OF HANGER ROD "D"	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$

NOTE:— FLOAT OR BUCKET TRAP SHOWN. IF THERMOSTATIC TRAP IS USED PROVIDE 5'-0" COOLING LEG BETWEEN TRAP AND UNIT HEATER



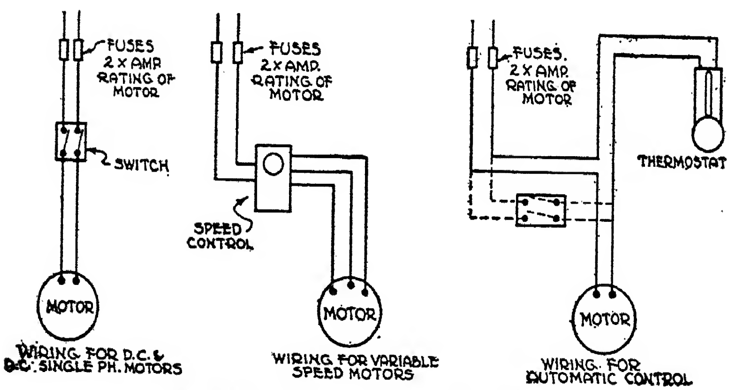


Fig. 145. Unit Heater Piping and Wiring Diagrams

outside temperature of 0°F. and inside temperature of 65°F. The wind pressure is 15 miles per hour. Assume 5 pound steam pressure at heaters.

(a) Calculate the total heat loss in B.t.u. per hour. (b) Select the number and size of heaters from Table 86 and explain fully why you made your

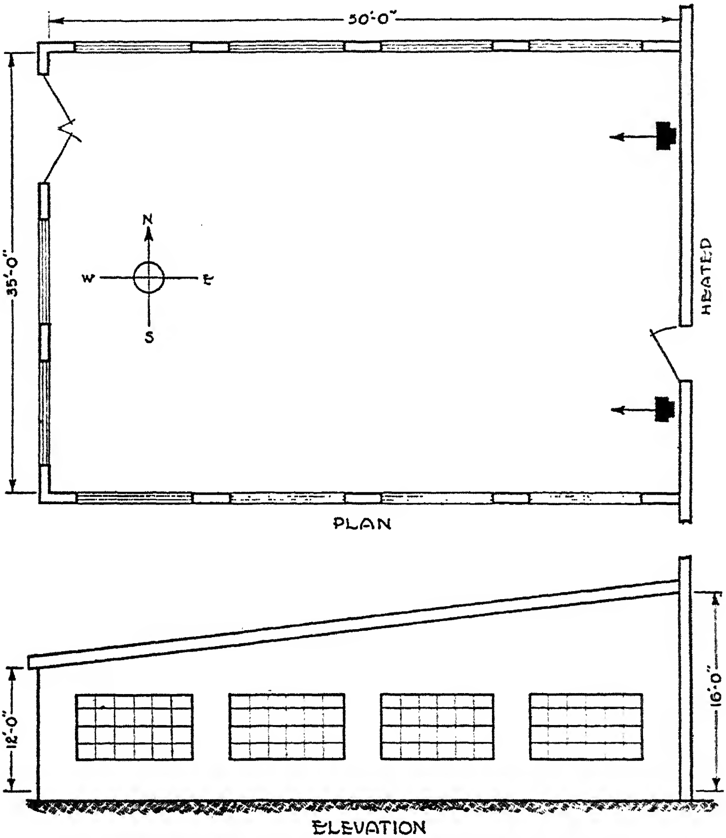


Fig. 146. Plan and Elevation View of a Typical Factory Building

selections. (c) Draw a rough sketch and show locations of the heaters. Tell why you located them at the points you select. (d) Explain how you would control the system. (e) What size boiler will be needed? (f) How could summer cooling be carried on in the factory without too much change in apparatus?

Gas Unit Heaters. Gas unit heaters, because they are so much different in operation, are here taken up as a separate type of heater. The gas unit heater requires no steam boiler, steam or water

lines, coal or other fuel storage, or firemen to operate it. The unit is complete in itself and needs only a gas supply line and electric current for its operation.

Fig. 147 shows a typical gas unit heater and Fig. 148 shows typical details and dimensions for three typical sizes of units. It

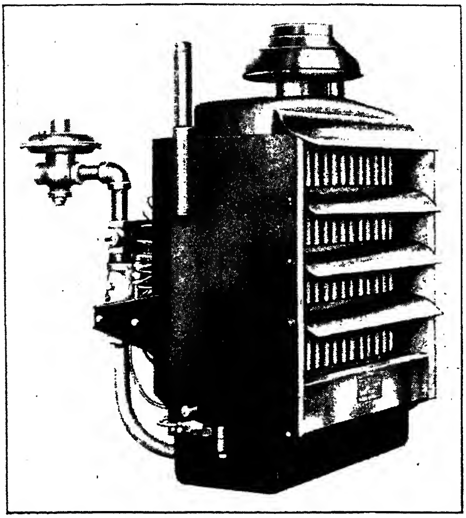
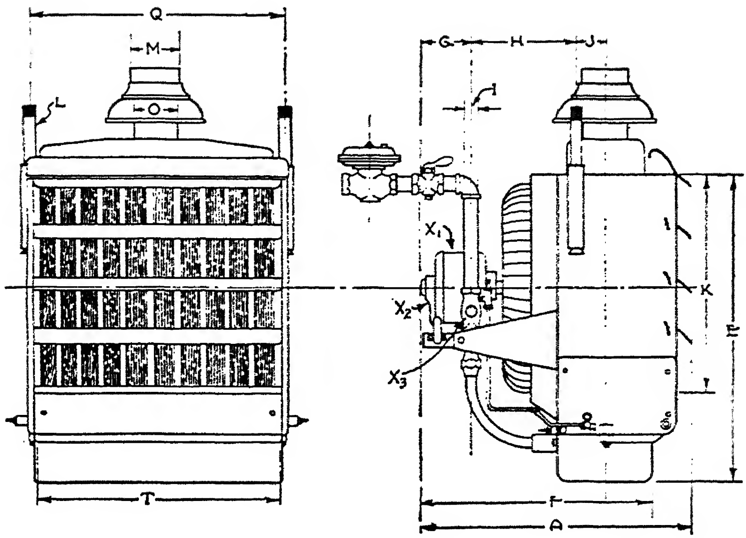


Fig. 147. Typical Gas Unit Heater
Courtesy of General Gas Light Company, Kalamazoo, Michigan

will be noted that the operation is on the same principle as steam or electric units insofar as louvers and fans are concerned. The fan forces air around the coils and the louvers direct its path. The units can be suspended from ceilings, hung from walls, or built up from the floor depending on the requirements. When it is necessary to diffuse the air over a large open area or direct it in one particular direction, the units are suspended from the ceiling. In such a position they are out of the way and the louvers can be adjusted to send the air in any required direction. Sometimes it is desirable to heat more than one (small) room with a single heater. This can be done with a single duct system connected to the face of the unit by a transition piece which reduces the face area to the duct



	No. 60 Inches	No. 75 Inches	No. 100 Inches
A	24 1/8	24 1/8	25 7/8
B	24 1/2	26 1/2	30 7/8
C	21 1/8	21 1/8	26 3/8
D	17 1/16	17 1/16	22 7/8
E	21 1/4	23 1/2	26 7/8
F	21 1/8	21 1/8	22 5/8
G	6	5	5 1/2
H	10 7/8	10 7/8	10 1/8
I	3/4	3/4	1
J	1 7/8	1 7/8	3
K	14 1/8	16 3/8	18 5/8
L	3/4	3/4	1
M	7	9 3/8	9 3/8
N	4	5	5
O	3	4	4
P	6 1/2	6 1/2	6 1/2
Q	18 7/8	18 7/8	24 7/8
R	3 3/4	3 3/4	6 3/4
S	4 1/8	4 1/8	6 3/4
T	16 3/4	16 3/4	22 5/8

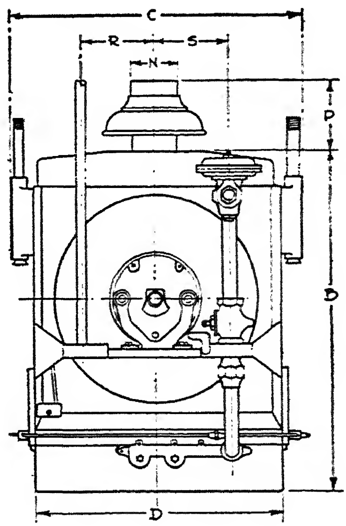


Fig. 148. Typical Details and Dimensions for Gas Unit Heaters
Courtesy of General Gas Light Co., Kalamazoo, Michigan

area. (See Fig. 149.) When used in this manner the unit is generally hung on a wall in an out of the way location. It should be remembered that the unit must be operated only with a minimum of static pressure and for this reason the duct should not be long or contain

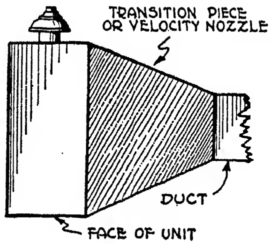


Fig. 149. Typical Use of Velocity Nozzle Attachment When Unit Is Used in Connection with a Duct System

sharp turns. In still other cases where rooms have very high ceilings, the unit is suspended from the ceiling but instead of directing the air by means of louvers a 90° transition piece is employed which directs the air downward as it leaves the unit. (See Fig. 150.)

Table 87 shows typical specifications for several sizes of units, together with capacities, fan speeds, etc.

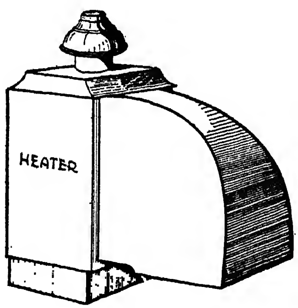


Fig. 150. Typical Use of Velocity Attachment When Unit Heater Is Used in Rooms Having Very High Ceilings

Installation. The proper location of gas unit heaters must be considered carefully if best results are to be obtained. Due to varying conditions it is impossible to make arbitrary rules govern-

ing the placing of units; however, certain factors must be recognized.

If more than one gas unit heater is used in the same area, each unit should be so spaced that it will distribute heat in a definite portion of the building. The units should be arranged in such relation to each other that a continuous circulation of the heated air will be maintained in the room.

Conflicting currents of warm air are to be avoided. In heating a building with large glass exposure areas, it is often good practice to spot the heaters so that the heated air brushes the glass at an angle. This immediately warms any cold air filtering in and prevents radiation of heat toward these colder areas.

In buildings where doors are opened frequently, heat should be directed toward the doorways to combat the inrush of cold air.

***Table 87. Typical Specifications and Capacities for Gas Unit Heaters**

Number	60	60	75	75	100	100	100
Input Rating B.T.U. per hour	60,000	60,000	75,000	75,000	100,000	100,000	100,000
Fan Speed R.P.M.	860	1140	860	1140	685	860	1140
Fan Diameter	14 15/16"	12½"	14 15/16"	14 15/16"	18⅞"	19"	19"
No. of Blades	4	6	4	4	6	6	6
Pitch	2"	2"	2⅜"	2"	2½"	1¾"	1⅛"
Motor Horsepower	1/20	1/20	1/20	1/15	1/20	1/15	.1/6
Air Velocity F.P.M. 1 Ft. from Unit	565	535	510	665	515	700	800
Air Delivery C.F.M.	865	825	1000	1235	1450	1835	2150
Temperature Rise °F.	50°	54°	56°	46°	54°	43°	40°
Output Equivalent in Sq. Ft. Steam Radiation	207	207	259	259	346	346	346
Net Weight	150	150	158	158	232	232	232
Shipping Weight	200	200	230	230	360	360	360

*Heaters operate 83 per cent efficient when vented.

There are two general methods of hanging a gas unit heater—by suspension from the ceiling and by suspension from a wall bracket.

When suspending the unit from the ceiling, it is always advisable to install the unit at a height from eight to twelve feet from the floor. Located in this position and by directing the adjustable louvers on the front of the heater slightly downward, warm air that naturally rises to the ceiling, where heat loss is usually great, will be drawn through the unit and forced down to the lower working level.

A unit should be installed at least 12 inches from the ceiling to facilitate easy vent pipe connections, where venting is found necessary.

A wall bracket installation is desirable in the case of exceptionally high ceilings, where it is difficult to make a ceiling suspension installation. A strong bracket with firm supports can be easily constructed from either pipe or angle iron.

Table 88. Pipe Size Chart

Showing Capacity of Pipe of Different Diameters and Lengths in Cu. Ft. per Hour with Pressure Drop of 0.2 In. and Specific Gravity 0.60

To Be Used for Figuring Laterals and Service Pipes

Length of Pipe Ft.	Diameter of Pipe, Inches						
	¾	1	1¼	1½	2	3	4
15	168	350	620	960	2000	5400	11200
30	120	245	430	680	1400	3800	7900
45	98	200	355	530	1150	3200	6500
60	84	175	310	480	1000	2700	5600
75	76	155	275	430	890	2450	5000
90	70	145	250	395	810	2260	4550
105	64	132	232	370	750	2100	4200
120	60	125	215	340	700	1950	4000
150	54	110	195	310	630	1750	3550
180	49	100	175	280	570	1600	3200
210	43	94	165	260	530	1450	3000

Gas and Electric Connections. After a gas unit heater is hung in position, gas connection is relatively simple. The gas line to the unit should be as short as possible. The size of the gas line that should be run to the various sizes of unit heaters is shown in Table 88. The hourly gas consumption for the various sizes of unit heaters is shown in Table 89. These figures are based on 1,000 B.t.u. natural gas and 500 B.t.u. manufactured gas. Data relative to various types of gas used in numerous localities can be secured from local Gas Companies.

A gas pressure regulator is furnished with every gas unit heater to maintain a uniform pressure at the burner. When burning natural gas, a regulator is furnished to maintain a 5-inch pressure at the burner. When burning manufactured gas, a gas regulator to maintain a 3-inch pressure is furnished.

The regulator should be installed in the gas line about 18 inches from the gas valve. Care should be taken in installing it in a horizontal position with the correct side up and with gas flowing in the direction indicated by the arrow on the regulator. Read instructions on the regulator carefully before making installation to insure proper functioning of the apparatus.

Note: If regulator has shipping pin in position make sure this is removed before regulator is installed.

A hand valve should be connected in the gas supply line at the rear of every unit heater installed. This valve should remain in a closed position until the unit heater is ready to be started.

To make unit heater electrical connections, a conduit pipe or B-X carrying No. 14 wire should be run to the conduit box which is at the left of the motor at the rear of the heater. All electrical connections should conform with Standard Building Codes. Units are equipped for either 110- or 220-volt A.C. single-phase motors, any cycle; or D.C. 115- or 230-volt motors.

Table 89. Hourly Gas Consumption Chart

Gas Unit Heater	Hourly B.t.u. In-put	Mfg. Gas Hourly Consumption	Nat. Gas Hourly Consumption
60	60,000	120 cubic feet	60 cubic feet
75	75,000	150 cubic feet	75 cubic feet
100	100,000	200 cubic feet	100 cubic feet
125	125,000	250 cubic feet	125 cubic feet
200	200,000	400 cubic feet	200 cubic feet

Operation. The operation of the gas unit heater is simple. When motor *X1*, Fig. 148, is started, the governor on fan shaft opens by centrifugal force permitting rocker arm *X2* to release plunger in main gas valve *X3*. This opens the valve and supplies gas to the main burners.

Heat from the main burners is communicated to the radiating elements above the burners. Air is delivered by fan across the elements, absorbs the heat and forces the warm air out into the room. A definite volume of air is directed into the burner box assuring complete combustion.

The products of combustion pass through the radiating elements to the vent housing and can be easily disposed of through a vent pipe.

The radiating elements are so arranged as to allow the air from the fan to pass over them with the minimum of air resistance and to eliminate eddy currents. The unit delivers a large volume of air at a low temperature rise which assures an even distribution of heat throughout the room.

Venting. Fig. 151 illustrates a typical vent system for a gas unit heater. The figure is self-explaining.

Although the units when unvented do not discharge dangerous gases into the building, a flue should always be used to direct the

products of combustion, if not out of doors at least into a room away from any combustible materials. An effective method of venting a gas unit heater is shown in Fig. 151. Where it is necessary to run long lines for flues, the matter of handling the condensation problem within the flue or chimney resolves itself into either

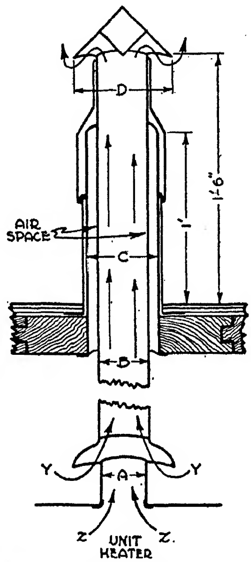


Fig. 151. Typical Vent System for a Gas Unit Heater

preventing condensation or a satisfactory disposal of it after formation. There are three general methods of preventing it.

(1) Make flue pipe as short and as direct as possible.

(2) Keep pipe within heated area of building, away from cold surfaces wherever possible.

(3) Insulate pipe to maintain high temperature in pipe.

General provisions for satisfactory disposal of condensate where it is not easily prevented:

(1) Pitch flue pipe slightly downward toward properly lined chimney and provide drain at base of chimney if large amount of condensation is present.

(2) Installation of drain attached to low point in flue pipe.

Example. It is required to make a heating estimate for a one-story garage building in Detroit, Michigan. The building is 100 feet long, 50 feet wide, and 16 feet high. (See Fig. 152.) The floor is constructed of 6-inch concrete on the ground. The roof of the garage proper is of 2-inch planking covered with roofing. The office roof is of the same construction with a wood

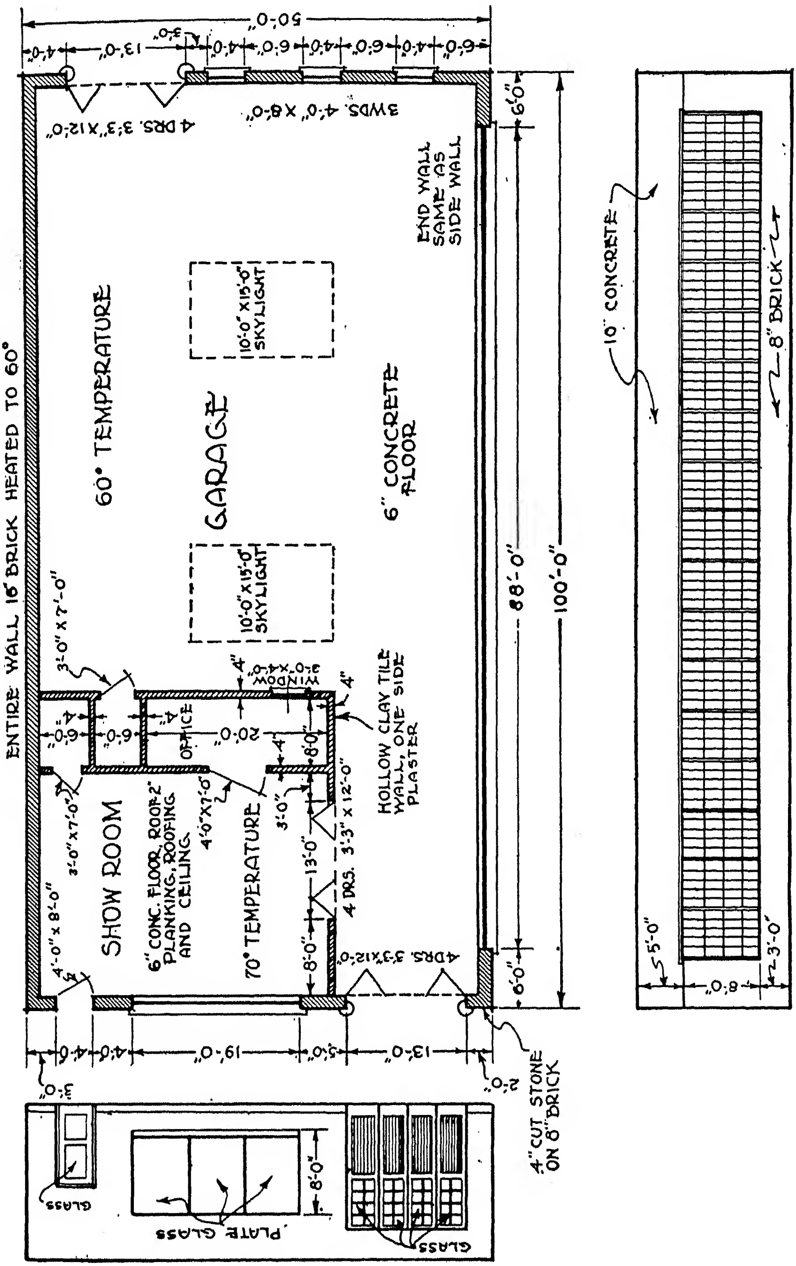


Fig. 152. Typical One-Story Garage Building

ceiling. Three walls of the building are exposed. One side wall is 16-inch brick, and adjoins a building heated to 60 degrees. The back wall and exposed side wall are constructed of 10-inch concrete, 5 feet up from the ground, and the balance of the wall is 8-inch brick. The front wall is constructed of 4-inch cut stone on 8-inch brick. There are two 10x15 skylights in the garage, shown in Fig. 152.

The 70 degree temperature is to be maintained in the office during working hours and 50 degrees during non-working hours. The garage temperature is to be 60 degrees during working hours and 50 degrees during non-working hours. The minimum outside temperature is figured at 0°F. The number of working hours per week in the garage is ten hours a day for five days, and

Table 90. Heat Loss Calculations

Due to the fact that the temperature to be maintained in the office is different from that of the shop, it is necessary to figure two separate hourly heat loss estimates.

Heat loss of garage proper

Temperatures 0°F. to 60°F.

A Building Material	*B* Area or Volume[1]		*C* Coeff. of Trans.[2]		*D* Temp. Diff.		*E* B.t.u. Loss
Glass	800	×	1.13	×	60	=	54,240
Door ½ Glass	312	×	1.13	×	60	=	21,154
Net Wall 8″ Brick	759	×	.50	×	60	=	22,770
Floor 6″ Concrete	3,912	×	.90	×	10	=	35,208
Net Roof 2″ Plank and Roofing	3,612	×	.32	×	62	=	71,662
*Air Change 1	62,592	×	1.1	×	..	=	68,851
Skylight	300	×	1.13	×	62	=	21,018
Net Wall—10″ Concrete	685	×	.63	×	60	=	25,482
Net Wall—On 8″ Brick, and 4″ Cut Stone	100	×	.37	×	60	=	2,220
Total						=	322,605
Heat supplied through wall exposure of office[3]						=	4,145
							318,460

No. Heaters Required $\frac{318,460}{83,000^4}$ =3.84 or 4 No. 100 Units or 2 No. 100 Units and 1 No. 200 Unit.[4]

[1]Areas figured from Fig. 152.
[2]Coefficients are calculated by using Formulas (2) to (7) or by locating them in Tables 2 to 13 in Vol. II.
[3]This is calculated by using Formula 1, Vol. II, and assuming temperature difference as 70°—60° or 10°F.
[4]Explained below.

The first thing to do in figuring the hourly heat loss of a building is to list the building material as shown in Column A of Table 90. After all the building material is listed, figure out the exposed surfaces or area of this material as shown in Column B. By referring to the dimensions on the drawing, areas can be easily calculated.

Next, the coefficients of transmission of heat for the building construction material should be listed as shown in Column C. By referring to Tables 2 to 13, Vol. II, the coefficients of transmission for most building materials can be found.

Building Material	Coefficients of Transmission
Glass	1.13
Door	1.13
Net Wall 8″ Brick	.50
Floor 6″ Concrete	.90
Roof 2″ Plank-Roofing	.32
Skylight	1.13
Net Wall—10″ Concrete	.62
Net Wall—4″ Cut Stone on 8″ Brick	.37

*In most cases, one air change should be figured. However, in some cases where doors are frequently open, or where there are exhaust fans, 2 or maybe 3 air changes should be figured, all depending upon the severity of the condition. In this case, 1 air change is figured. To find the coefficient for air changes, up to 3 changes per hour, refer to air change chart. See Table 90A. The air change for a 60 degree temperature difference is 1.1.

To find hourly heat loss through air change, multiply the volume of air in the building by the air change coefficient. In the case of the garage, multiply 62,592 of Column B by 1.1 of Column C. The hourly B.t.u. heat loss through air change in this case is 68,851 B.t.u.

five hours for one day (Saturday); in the office ten hours a day for six days. The price of gas per one thousand cubic feet is assumed to be sixty-five cents. The heat value of the gas is assumed to be 550 B.t.u.

Note: It is assumed that the reader is acquainted with the material given in Vol. II, Chapters V and VI relative to calculation of coefficients and heating loads. This sample estimate goes about the calculation of the heating loads or heat losses, using a different form of the B.t.u. method than was explained earlier in this book. However, this deviation is a typical one, and it is felt that the reader should have experience in other than this form, because many engineers use other than the usual forms. The calculations should be easily understood.

The temperature difference is the difference between the minimum outside and maximum inside temperature. The outside temperature in this case was taken at zero, therefore in this garage the temperature difference of the exposed surfaces is 60 degrees. In attics, unheated basements, and similar enclosures, the temperature of the enclosure is assumed to be one-half the difference between the outside and inside temperature. In case there is a basement under the garage, the temperature difference of the floor would be 30 degrees. In case the floor is on the ground, the ground temperature is assumed to be 50 degrees. Therefore the temperature difference in this case is 10 degrees in the garage and 20 degrees in the office. Where ceilings or roofs are over 12 feet, there is also a variation. For

Table 90A. Air Change Chart

B.T.U. Per Cubic Foot of Room Contents

ONE AIR CHANGE PER HOUR

Design Temperature Wt. per cu. ft.	Final Temperature				
	40°F. .0795 lbs.	50°F. .0779 lbs.	60°F. .0764 lbs.	70°F. .0750 lbs.	80°F. .0735 lbs.
−10°F.	0.955	1.122	1.282	1.440	1.590
0	0.763	0.935	1.100	1.260	1.411
+10	0.573	0.748	0.927	1.080	1.235
+20	0.382	0.561	0.734	.900	1.060
+30	0.191	0.374	0.550	.720	.883

TWO AIR CHANGES PER HOUR

Design Temperature Wt. per cu. ft.	Final Temperature				
	40°F. .0795 lbs.	50°F. .0779 lbs.	60°F. .0764 lbs.	70°F. .0750 lbs.	80°F. .0735 lbs.
−10°F.	1.190	2.244	2.564	2.880	3.180
0	1.526	1.870	2.200	2.520	2.822
+10	1.146	1.496	1.854	2.160	2.470
+20	0.764	1.122	1.468	1.800	2.120
+30	0.382	0.748	1.100	1.440	1.766

THREE AIR CHANGES PER HOUR

Design Temperature Wt. per cu. ft.	Final Temperature				
	40°F. .0795 lbs.	50°F. .0779 lbs.	60°F. .0764 lbs.	70°F. .0750 lbs.	80°F. .0735 lbs.
−10°F.	2.865	3.366	3.846	4.320	4.770
0	2.289	2.805	3.300	3.780	4.233
+10	1.719	2.244	2.781	3.240	3.705
+20	1.146	1.683	2.202	2.700	3.180
+30	0.573	1.122	1.650	2.160	2.649

Table 91. Heat Loss Calculations

Heat loss of garage office and showroom:

Temperatures 0° to 70°.

A Building Material	*B* Area or Volume		*C* Coeff. of Trans.		*D* Temp. Diff.		*E* B.t.u. Loss
Glass	152	×	1.13	×	70	=	12,023
Door—All Glass	32	×	1.13	×	70	=	2,531
Net Wall—4″ Cut Stone on 8″ Brick	360	×	.37	×	70	=	9,324
Floor 6″ Concrete	1,088	×	.90	×	20	=	19,584
Net Roof—2″ Plank Roofing and Ceiling	1,088	×	.24	×	72	=	18,800
Air Change—1	17,408	×	1.26	×	..	=	21,934
Net Wall—16″ Brick (Adjoins heated building)	512	×	.27	×	10	=	1,382
*Door—All Wood (Adjoins heated room)	177	×	.46	×	10	=	814
*Window (Adjoins heated room)	12	×	1.13	×	10	=	135
*Net Wall—4″ Hollow Clay Tile—One Side Plaster (Adjoins heated room)	869	×	.42	×	10	=	3,650
Total						=	90,177

No. Heaters Required $\frac{90,177}{83,000} = 1.085$.

*Heat loss through these areas (4,145 B.t.u.) of the office and showroom should be credited toward heating the garage proper.

the first 12 feet of ceiling height, the room temperature is figured regularly. Every ten feet above the first 12 feet, add 4 degrees to room temperature difference. The temperature difference for the ceiling of this garage is figured as 62 degrees as the ceiling is 16 feet high. By multiplying the area of the building material, by the coefficient of transmission of this material, by the temperature difference of the material, the hourly B.t.u. heat loss for the material is found, which should be listed as shown in Column E.

To find the hourly heat loss of the room, add Column E. In the case of the garage, the total hourly B.t.u. loss is 322,605, less 4,145 which is supplied through wall exposure of the office. This leaves a heat loss of 318,460 B.t.u. for the garage.

To find the number of heaters to heat the room, divide the total hourly B.t.u. heat loss of the room by the hourly B.t.u. output of the gas Unit Heater. The hourly B.t.u. input of a No. 100 Humphrey Gas Unit Heater is 100,000 B.t.u.; the hourly B.t.u. output of this Unit is 83,000 B.t.u. because it operates 83% efficient when vented. In the case of the garage, divide 318,460 by 83,000 which equals 3.84 or 4 Unit Heaters necessary to heat the room to 60 degrees with an outside temperature of zero degrees.

When there are two rooms under the same roof adjoining each other and a lower temperature is maintained in one than in the other, it is important to note the infiltration of heat through the walls from the room of a higher temperature to the room of a lower temperature, as well as to note the heat loss of the room of the higher through the walls to the room of a lower temperature.

In the case of the garage building, the infiltration from the office and showroom through the wall to the garage proper is 4,145 B.t.u. per hour. This figure should be subtracted from the hourly heat loss of the garage proper.

The hourly heat loss for the garage office and showroom is figured in the same manner as the garage proper but of course using different temperature difference, building materials, etc.

Figure Gas Consumption and Operating Cost for the Heating Season. The first thing to do in making fuel requirement calculations is to figure the average degree hours per day for each zero degree day at the temperature the room is to be heated.

The Garage proper is to be heated:

*12 hours at 60° per day for 5 days per week
°12 hours at 50° per day for 5 days per week
* 6 hours at 60° per day for 1 day per week
°18 hours at 50° per day for 1 day per week
°24 hours at 50° per day for 1 day per week

In most cases, it is desirable to maintain a lower temperature during non-working hours than during working hours. In the case of the garage proper, the temperature is dropped 10 degrees during non-working hours.

When a lower temperature is to be maintained, one hour should be added to the time the working hour temperature is maintained to allow for pickup. In the case of the garage proper, the building is heated to 60 degrees during working hours, which is 10 hours per day—one hour should be allowed for heating during noon hour and one hour for pickup to bring the non-working hour temperature of 50 degrees up to 60 degrees, the working temperature desired. Thus, the total heating hours for a 10 hour working day is 12 hours.

In cases where a lower temperature is maintained, it is necessary to make two calculations for the average degree hours per day for each zero degree day, one for the working hour temperature and one for the lower non-working hour temperature.

Working hour temperature:

*$12 \times 60 \times 5 = 3{,}600$
* $6 \times 60 \times 1 = 360$
$3{,}960$

$$\frac{\text{Degree hours}}{\text{Temperature of room during working hours} \times \text{days in period}} = \text{Average Degree}$$

Hours per day for each zero degree day at temperature of room during working hours.

or,

$$\frac{3{,}960}{60 \times 7} = 9.43$$

Non-working hour temperature:

°$12 \times 50 \times 5 = 3{,}000$
°$18 \times 50 \times 1 = 900$
°$24 \times 50 \times 1 = 1{,}200$
$5{,}100$

$$\frac{\text{Degree hours}}{\text{Temperature of room during non-working hours} \times \text{days in period}} = \text{Average De-}$$

gree Hours per day for each zero degree day at temperature of room during non-working hours.

or,

$$\frac{5{,}100}{50 \times 7} = 14.57$$

All data so far obtained should be listed as follows:

B.t.u. heat loss per hour of garage proper		318,460
*Degree Days	Working hour at 60°	4,089
	Non-Working hour at 50°	2,240
Average Degree hours per day	Working hour at 60°	9.43
	Non-Working hour at 50°	14.57
B.t.u. per cu. ft. of gas		550
Efficiency of Unit Heater when vented		83%
Price of gas per thousand cu. ft.		65c

*To find degree days, refer to degree day chart, Vol. II. Degree day temperatures should be figured five degrees lower than the temperature used in calculating heat loss, for a mean temperature of 65 degrees denotes a daytime temperature of 70 degrees, 55 degrees denotes a daytime temperature of 60 degrees, and 45 degrees denotes a daytime temperature of 50 degrees. If you are figuring an estimate for a town not listed in the degree day chart, select a nearby town that is listed.

The equation for finding the gas consumption per season is as follows:

$$\frac{\text{Maximum hourly heat loss} \times \text{Degree days} \times \text{Average degree hours}}{\text{B.t.u. value of gas} \times \text{Efficiency of unit} \times \text{Room temperature less 5 degrees}} = \text{Cu. ft. per season.}$$

Annual gas consumption during working hours:

Room temperature 60 degrees.

$$\frac{318{,}460 \times 4{,}089 \times 9.43}{550 \times .83 \times 55} = 489{,}080 \text{ cu. ft.}$$

Annual gas consumption during non-working hours:

Room temperature 50 degrees.

$$\frac{318{,}460 \times 2{,}240 \times 14.57}{550 \times .83 \times 45} = 505{,}952 \text{ cu. ft.}$$

Total gas consumption for room per season:

$$489{,}080 + 505{,}952 = 995{,}032 \text{ cu. ft.}$$

To estimate cost of gas per season:

$$\frac{\text{Cu. ft. of gas per season}}{1000} \times \text{price of gas per 1000 cu. ft.} = \text{Cost of gas per season.}$$

$$\frac{995{,}032}{1000} \times .65 = \$646.77 \text{ cost of gas per season.}$$

The same plan is followed in figuring the gas consumption per season and season's operating cost for the garage office and showroom.

The garage office and showroom are to be heated:

*12 hours at 70° per day for 6 days per week.
°12 hours at 50° per day for 6 days per week.
°24 hours at 50° per day for 1 day per week.

Working hour temperature:

$$*12 \times 70 \times 6 = 5{,}040$$

$$\frac{5{,}040}{70 \times 7} = 10.29$$ The average degree hours per day for each degree day at 70°.

Non-working hour temperature:

$$°12 \times 50 \times 6 = 3{,}600$$
$$°24 \times 50 \times 1 = 1{,}200$$
$$4{,}800$$

$\frac{4{,}800}{50 \times 7} = 13.71$ The average degree hours per day for each degree day at 50°.

Listing data:

B.t.u. heat loss per hour of garage office and showroom	90,177
Degree Days {Working hour} at 70°	6,494
Degree Days {Non-working hour} at 50°	2,240
Average Degree Hours per day {Working hour} at 70°	10.29
Average Degree Hours per day {Non-working hour} at 50°	13.71
B.t.u. per cu. ft. of gas	550
Efficiency of Unit Heater when vented	83%
Price of gas per thousand cu. ft	65c

Figuring equation:

Annual gas consumption during working hours:

Room temperature 70 degrees.

$$\frac{90{,}177 \times 6{,}494 \times 10.29}{550 \times .83 \times 65} = 203{,}081 \text{ cu. ft.}$$

Annual gas consumption during non-working hours:

Room temperature 50 degrees.

$$\frac{90{,}177 \times 2{,}240 \times 13.71}{550 \times .83 \times 45} = 134{,}812 \text{ cu. ft.}$$

Total gas consumption for room per season:

$$203{,}081 + 134{,}812 = 337{,}893 \text{ cu. ft.}$$

Estimated cost of gas per season:

$$\frac{337{,}893}{1000} \times .65 = \$219.63$$

Refer to Fig. 153 for recommended location of heaters.

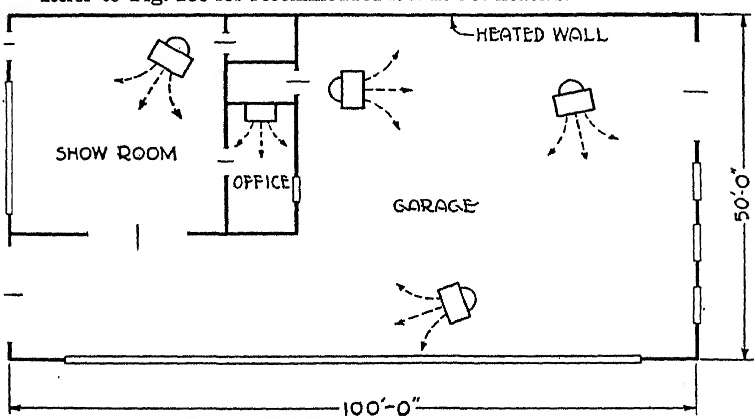


Fig. 153. Shows Recommended Location of Heaters

SUMMARY

Hourly B.t.u. heat loss of garage proper		318,460
Hourly B.t.u. heat loss of garage office		90,177
No. of Humphrey Gas Unit Heaters to heat garage proper	2 No. 100, with 1150 r p.m. fan 1 No. 200, with 1140 r.p.m. fan	3
No. of Humphrey Gas Unit Heaters to heat garage office and showroom	1 No. 100, with 685 r.p.m. fan 1 No. 20 Ambassador	2
Estimated gas consumption per season for garage proper		995,032
Estimated gas consumption per season for garage office and showroom		337,893
Total estimated gas consumption per season for Detroit garage building		1,332,925
Estimated cost of gas per season for garage proper		$646.77
Estimated cost of gas per season for garage office and showroom		$219.63
Total cost of gas per season for Detroit garage building		$866.40

CHAPTER XII

AUTOMATIC CONTROLS

Automatic control in connection with ordinary heating methods and air conditioning is a subject of prime importance in modern heating systems or air-conditioning units, because without it no system will operate efficiently. Automatic control provides not only comfortable and reliable conditions but it also brings about a much desired fuel economy. It is therefore important that the principles of automatic control be well understood. The following material outlines the subject first, as to purposes and definitions and finally, as to individual instruments and system operations.

Purposes of Control. The control system may be adapted to accomplish various functions. Any completely controlled system will include control devices which, when properly interlocked and coordinated, will provide for any or all of the functions listed herein. Due to the wide variance in the types of systems in terms of results to be produced, the functions of control assume variable importance from one system to another.

Most forms of automatic control are applied to heating, ventilating or air-conditioning systems to insure the maintenance of certain desired or required conditions of temperature, pressure, humidity, air motion or air distribution. This does not imply that these conditions are always fixed, as the most recent practice indicates the desirability of automatic control which will permit certain fluctuations in the individual factors so long as the final results in terms of comfort, efficiency or cost of operation are dependably and accurately maintained.

Control devices are often applied purely to serve a safety function, limiting pressures and temperatures within predetermined maxima or minima. In other instances, equipment must be started or stopped in definite sequence in order that it function without difficulty and without hazard. As an example, the gas burner must not be permitted to feed gas into the combustion chamber of a boiler or furnace unless the pilot light for its ignition is in proper working

order. Likewise, the vacuum refrigeration machine must not be permitted to operate until definite steam pressures are available for the jets and unless the various necessary pumps are placed in operation.

Elimination of human error is another purpose accomplished by the use of automatic control. While, from the standpoint of performing certain operations, it is physically possible for human operators to regulate heating, ventilating or air-conditioning systems, there is always the possibility of error or neglect which may be definitely detrimental, if not actually dangerous, in terms of results. Automatic controls reacting to definite physical conditions eliminate the possibility of error or neglect.

Modern equipment provided for the automatic control or regulation of heating, ventilating or air-conditioning systems has been so carefully and accurately designed that these devices are, without question, an improvement on human capabilities of performing similar actions. For instance, temperature and humidity sensitive devices will react to changes in temperature or humidity conditions before these changes become apparent to manual operators. It, therefore, follows that the results of regulation are more accurate and dependable with automatic control than with manual control.

While in the simpler forms of systems the cost of manual operation in terms of man-hours of labor is small, the cost of such regulation increases rapidly with the size and complexity of the system. It is similarly true that automatic control is more expensive on the larger and more complex systems, but in the case of automatic control this expense is primarily in the original or first cost of the equipment, whereas the manual regulation becomes a permanent item to be added to the general cost of operation of the system.

Due to the ability of automatic control to interlock and coordinate the various functions of the complete system in a manner difficult to accomplish with manual regulation, it is a generally accepted fact that a system may be operated at lower cost when automatically controlled. This lower cost regulation is due principally to the ability of control devices to maintain the operation of the system within the limits established and in the ability of the control devices to react to changing conditions without delay and without costly failures.

Particularly in the residential type of installation, the ability of automatic control to take the place of human attention is a factor not to be overlooked. While it may be said that automatic control in such instances is used to satisfy human indolence, there is no gainsaying the fact that the ability of automatic control to insure correct and dependable results with a minimum of attention is an item of major importance. While this factor is of less importance in commercial, theatre, office building, industrial applications, etc., it is still a factor of interest and importance to the purchaser to know that such devices will insure direct results without responsibility on the part of himself or his operators.

Glossary of Terms. Following are given a few of the most used terms and their usual definitions.

Thermostats. Thermostats will be defined as temperature sensitive devices reacting to air temperatures within a room. Such a device is normally installed on the wall of the room whose temperature it is to control, and in reacting to rising or falling temperatures the thermostat causes the operation of heating or cooling equipment such that desired temperatures will be maintained.

Temperature Controllers. All devices, except room thermostats reacting to temperature changes are, for the purpose of this discussion defined as temperature controllers. In this group are included devices intended for the regulation of duct temperatures, tank temperatures, coil temperatures or for the measurement of outdoor temperatures. Ductstats, aquastats, furnacestats, and airstats are examples of this group.

Pressure Controllers. Pressure controllers are defined as devices reacting to pressure and pressure changes. Examples of such devices are the pressure controls governing the operation of refrigeration equipment from either head or suction pressure, devices reacting to steam or water pressure or the pressure of air in the distribution systems.

Damper Motors. Damper motors, as discussed herein, will be defined as specialized power units, the purpose of which is to position fresh air, face, by-pass or distribution dampers regulating the flow of air through the system. Connected by suitable linkage, these damper motors react at the command of thermostats, temperature controllers and pressure controllers in adjusting air flow to the needs of the system.

Control Valves. Control valves are defined as water valves or air valves which may be adjusted by or at the command of controllers to regulate the flow of the medium passing through them to the needs of the system. Such control valves are usually constructed with a power unit linked to the valve stem in such manner that movement of the power unit at the command of a thermostat or controller will react to position the valve as conditions demand.

Solenoid Valves. Solenoid valves are, as their name implies, valves actuated by the magnetic effect of an electric solenoid built within them. While normally these valves are opened when the solenoid is energized, they are sometimes built in a reverse acting manner and closed when energized. In

heating, ventilating, and air-conditioning systems they are normally adapted to the control of oil or gas burners as fuel valves, as water valves on the smaller humidifiers, or as refrigerant valves in refrigeration systems. Solenoid valves are also, often adapted as pilot or relay valves in the operation of large diaphragm valves.

Limit Controls. This term is applied to any controlling device, the primary function of which is to limit or prevent the operation of the system in such a manner that temperatures or pressures exceed predetermined limits. High and low limit controls function to control high and low temperatures or pressures.

Humidity Controls. Humidity controls are automatic devices reacting to changes in humidity. Normally such devices act upon changes in relative humidity. Within this group of controls the instruments commonly called *humidity controls* operate to prevent relative humidity from exceeding a predetermined maximum. When constructed in a reverse acting manner, these instruments are known as dehumidification controllers, the action of which regulates the equipment used for dehumidification.

Furnace Fan Controls. In the operation of forced warm air heating systems, it is usually necessary to use some devices to prevent circulation of air until a sufficiently high temperature is attained in the furnace to prevent delivery of cold air. Such a device or instrument is called a furnace fan control or furnacestat.

Two-Position Control System. This system is often referred to as the **on** and **off** or positive action control. As an example, a simple device which starts and stops an oil burner or a fan is called a two-position control.

Floating Control System. Floating control is the name applied to a control system in which a control valve or damper motor will, upon changing extreme limits of the thermostat contact settings, come to rest only when the temperature has stabilized between those limits.

Modulating Control. The modulating control system is designated as gradual or graduated acting control or proportioning control. These names are synonymous as applied to automatic control and are used to designate the type of system in which a control valve or damper motor modulates or proportions the flow of air, steam or water in reacting to changes of conditions at the controller. Modulating control therefore means gradual changes in place of simply **on** or **off** as defined under floating controls. In a system under modulating controls, fractional changes in temperature, for example, are accurately noted and additional heat or cold air supplied to prevent the more noticeable changes that occur in the floating control systems.

Electric Control Systems. In such control systems the primary medium utilized to provide for the operation is electricity, and the basic function of these controls consists of switching or otherwise adjusting electric circuits to govern electric motors, relays, or other apparatus.

Pneumatic Control Systems. In the pneumatic systems the primary medium utilized to provide for the operation is a medium of compressed air, the pressure of which is varied by the controlling devices. By means of leak ports or orifices, the pressure of the air varies in the branch lines and the changing pressures are utilized in air operating devices.

Common Type Thermostats. These are constructed with temperature sensitive operating mechanisms of two general types. Probably the most popular actuating mechanism of the room thermostat group, and for any of the temperature controller group, is bimetal. Bimetal, which may be formed in flat strips, spirals or helices, consists of two metals having different coefficients of expansion which are welded together. Under the influence of temperature changes, the expansion of one of these metals at a greater rate than the other causes bending of the associated strips and a resulting movement and power which may be utilized to actuate contacts in the electric controls or the variable orifices in the pneumatic types of controls.

A second type of actuating means is found in the wide variety of thermostats and temperature controllers which utilize some form of flexible diaphragm or corrugated metal bellows. The most common practice is to utilize a closed assembly involving some highly expansive or volatile liquid which vaporizes and condenses under rising and falling temperatures. The internal pressure built up through this vaporization and condensation reacts in movement of the flexible diaphragm and this movement and power may, in turn, be linked to electrical contacts or to adjustable air orifices. The corrugated metal bellows is a special variety of diaphragm in which thin wall metal tubing is pressed or rolled into a corrugated form, this then being closed and filled with volatile liquid to react in the same manner. The principal reason for the corrugated construction is the greater flexibility and greater movement permissible from this structure.

Bimetal Type. This type, Fig. 154, consists of a bimetallic actuating unit with two or three wire controls which primarily operate from room temperature to directly control line or low voltage heating or cooling equipment within the electrical rating of the mercury switches. Where the electrical load is in excess of the thermostat ratings, these instruments are conveniently used as a pilot control for magnetic starters to regulate any load for which starters are available. Thus they find application in the control of unit heater or cooler motors, oil burners, stokers, refrigeration compressors, solenoid valves, or similar equipment used in governing the operation of heating and cooling cycles as found in the following illustrations.

Such thermostats embody a spiral bimetallic actuating unit which expands and contracts as the surrounding air temperature rises and falls. This movement is transmitted to tilt a mercury switch which in turn controls the imposed electrical load. This simple, accurate, and dependable mechanism is mounted on a drawn metal base and housed in an attractive brushed bronze case equipped with a thermometer. A bottom extension lever permits any tempera-

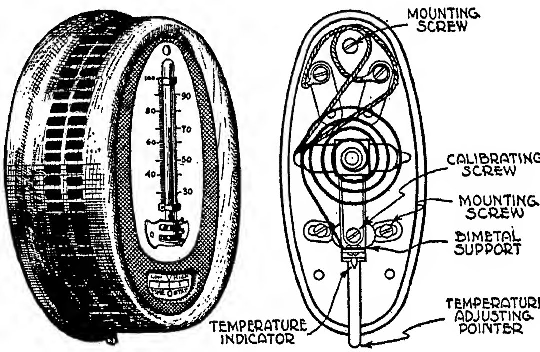


Fig. 154. Bimetal Type Thermostat. (This is obsolete but is shown here because many are still in use.)
Courtesy of Minneapolis Honeywell Regulator Company

ture setting within the range of the thermostat. The pointer and scale are visible through the front of the cover.

Bellows Type. This type, Fig. 155, consists of a bellows actuating unit with two or three wire controls which primarily operate from room temperature to a directly control line (or low) voltage heating or cooling equipment within the electrical rating of the mercury switches. Where the electrical load is in excess of the thermostat ratings, these instruments are conveniently used as a pilot control for magnetic starters to regulate any load for which starters are available. Thermostats find wide application in the control of unit heater or cooler motors, oil burners, stokers, refrigeration compressors, solenoid valves, or similar equipment used in governing the operation of heating and cooling cycles.

Certain types of thermostats embody a volatile fill bellows actuating unit which expands and contracts as the surrounding air temperature rises and falls. This movement is transmitted through a simple

lever and linkage arrangement to tilt a mercury switch which in turn controls the imposed electrical load. This simple, accurate, and dependable mechanism is mounted on a bakelite base and housed in an attractive, brushed bronze case equipped with a thermometer. A side extension knob permits any temperature setting within the

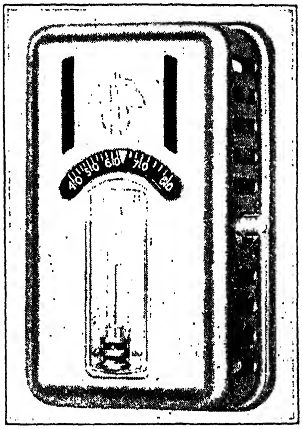


Fig. 155. Bellows Type Thermostat
Courtesy of Minneapolis Honeywell Regulator Company

range of the thermostat. The pointer and scale are visible through the front of the cover.

Clock Type Thermostats. Clock type thermostats are of three general types. They maintain a desired temperature throughout each day, and are so constructed, in conjunction with a clock, that they will move the thermostat temperature indicator to a lower night setting, thus greatly reducing fuel consumption. In the morning the clock automatically restores the desired daytime temperature setting. Probably the best known clock type thermostat is one making use of a spring clock.

A second type is operated in much the same manner as just described, except that the clock is electrically driven.

The third type uses either the spring or electric clock and in addition makes provisions for controlling temperatures in such cases where for one or more days, such as week-ends or holidays, the daytime required temperatures need not be the same as other days.

Eight-Day Type Thermostat. Fig. 156 shows a typical eight-day type thermostat designed to differentiate between daytime and nighttime temperatures.

Clock control of these types effects substantial fuel savings in the vast majority of installations. In addition, healthful and comfortable temperature regulation is effected. Medical authorities

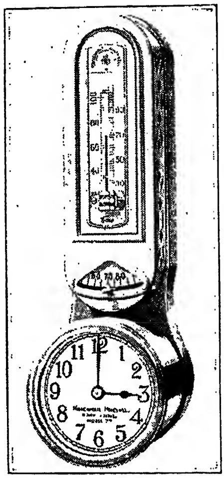


Fig. 156. Eight-Day Clock Type Thermostat
Courtesy of Minneapolis Honeywell Regulator Company

agree that lowering of the temperature at night to insure cool sleeping rooms is desirable.

The temperature control actuating means is a strip of two dissimilar metals, welded back to back into one piece and bent into a curved form. Since one of these metals expands and contracts faster than the other, rise in temperature tends to straighten the bimetallic strip, while reduction in temperature tends to curve it. One end of this bimetallic strip is firmly anchored to the center post of the thermostat. To the other end is fixed the contact carrying blade. This contact blade is thus moved by changes in room temperature, making and breaking the electrical control circuit, and thereby supplying more or less heat as required.

These thermostats employ open contacts and are designed only for low voltage operation. They possess the advantage of being calibrated along the entire scale range, permitting their use at any point on the scale without readjustment. Although strongly constructed, excessively heavy sections are avoided to prevent thermal inertia with its resulting slow thermostatic action. The location of the thermostatic element within the thermostat assures its intimate contact with the air, so it may follow temperature fluctuations most readily.

Ease of installation features all low voltage thermostats. The cable used passes easily through a small hole, eliminating any necessity of disfiguring the wall.

Electric Clock Type Thermostat. Fig. 157 shows a typical electric clock type of thermostat which operates on the same principle and to the same end as the eight-day type with the exception that the motive power is electrical instead of a spring-drive motor.

Week-End Type Thermostat. This type of thermostat may employ either the eight-day or electric clock principle and, in addition to controlling daytime and nighttime temperatures, it is designed to maintain what might be called nighttime temperatures during week-ends or other times when daytime temperatures are not required. This type is especially applicable to office buildings or other enclosures where the space need not be heated to the comfort point on Saturday afternoons and Sundays.

For brevity, the temperature desired to be maintained during working hours will be called the "Day Temperature," and the temperature which is desired to be maintained during the non-working hours will be called the "Night Temperature."

Example. The clock will raise the thermostat setting to the day temperature somewhat before the starting of the working day on Monday morning, and it will lower the setting to the night temperature at the close of the working day on Monday evening, and repeat this operation on Tuesday, Wednesday, Thursday, Friday, and Saturday, but on Sunday the clock will not move the temperature indicator and the setting will remain on night temperature from Saturday evening until Monday morning, or during the periods for which the week-end shutoff is set. Similarly, for a week day holiday, the thermostat will, by previous setting, continue the night temperature from the evening of the day before the holiday until the morning of the day after the holiday. The normal operation of the thermostat continues during the remainder of the week.

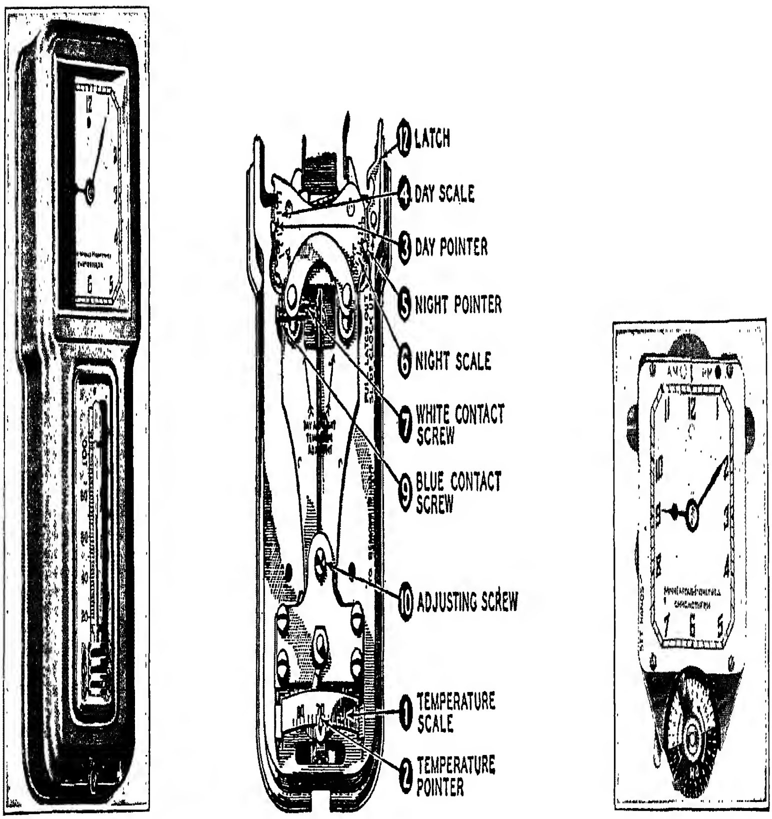


Fig. 157. Electric Clock Type Thermostat Showing, Left to Right, the Completely Assembled Thermostat, the Main Carriage, and the Electric Clock

Courtesy of Minneapolis Honeywell Regulator Company

There are several different styles of week-end or holiday clocks having shutoff dials for such fixed periods as follows:

From Saturday P.M. to Monday P.M.
From Friday P.M. to Monday A.M.
For multiple groups of days
For alternate days
Holiday in addition to week-end

Both the week-end and holiday shutoffs consist of 1-inch dials with filled in portions to retard the action of the trip hands, Fig. 157, and delay the change in temperature for a given length of time.

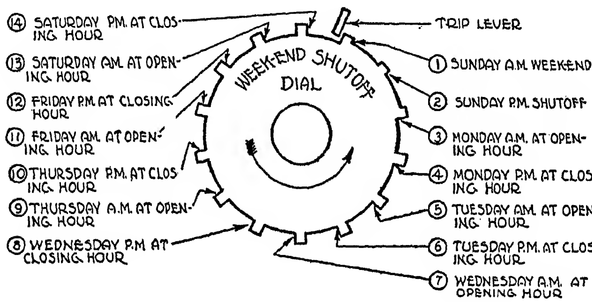


Fig. 158. Week-End Shutoff Dial for Chronotherms

The week-end shutoff has in addition to the filled in portion, 14 projecting pins—the space between each pin representing a half day operation of the thermostat. This means that although the 24-hour dial (Fig. 157) revolves once each 24 hours, the week-end shutoff dial revolves only once each seven days. The week-end shutoff is located under the 24-hour dial, Fig. 158. Due to the construction of the dial, the position of the filled in portion can be determined in reference to the day the instrument is being installed. For example, if an instrument is being installed on Tuesday morning after the change to normal daytime temperatures has taken place and the action of the week-end shutoff is desired from Saturday night to Monday morning, lift the trip lever, Fig. 158, and move the week-end shutoff dial until the trip lever drops down between the projecting pins 5 and 6, Figs. 158 and 159.

Refer to Figs. 158 and 159 and locate the filled in portion of the week-end shutoff being installed in relation to the filled in portion of the respective diagram. The correct location for the trip lever

between the projecting pins can be determined by applying the numbers, shown in Figs. 158 and 159, to the respective projecting pins of the week-end shutoff.

The holiday shutoff dial consists only of the filled in portion without projecting pins and is located between the week-end shutoff dial and the 24-hour dial. By depressing the lower week-end shutoff dial with thumbnail or knife, it will be found that the adjustable holiday dial may be moved to fill in any two of the cutout places desired without changing the position of the week-end shutoff dial.

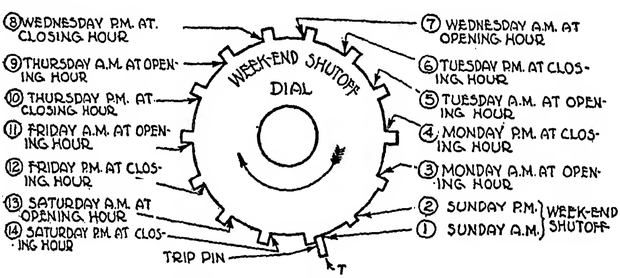


Fig. 159. Week-End Shutoff Dial for Clock Thermostat

Each cutout section corresponds to the half day of the week as indicated in Figs. 158 and 159. For example, if it is desired to use the holiday shutoff from Thursday morning until Friday morning, the filled in portion of the holiday shutoff should cover cutout places between pins 9, 10, and 11.

Special Type Thermostats. The special types are similar to the common types (see Fig. 157) with the exception that the special types have heat leveling or mechanical timing features.

The Chronotherm. The Chronotherm, as a typical example, is a sensitive electric clock (low voltage) room thermostat having either a mechanical timing or heat acceleration feature.

The mechanical timing type removes the inherent lag of common type thermostats by a feature which closes the contacts periodically, assuming constant air circulation and eliminating "Cold 70°." (This means a cold room even though the thermostat is at 70°F.)

The heat acceleration type offers the feature of not applying artificial heat until there has been a definite rise in room temperature.

The Chronotherm utilizes the electric current flowing through

the instrument, artificially heating the bimetal, shown in Fig. 154, and accelerating its normal action. This addition of artificial heat causes the bimetal to reach a temperature higher than the room temperature, thereby stopping the source of heat before sufficient time has elapsed to cause over-shooting. This insures efficient burner operation, definitely eliminates "Cold 70°" and improves the distribution of heat throughout the entire system.

The time necessary to produce a small increase in room temperature is dependent on the variables present at each burner operation, assuming gas or oil as fuel. A poorly insulated home would require a longer time for the temperature to reverse and start to rise instead of fall, following the start of the burner, than a well insulated one. Likewise, on a mild day the time will be shorter than on a cold day, and an over-sized heating plant will raise the temperature faster than an under-sized plant. Heat acceleration, therefore, assures a definite rise in room temperature during each operation of the heating plant, because the artificial heat will not be applied to the bimetal element until a definite rise in room temperature has been noted by the Chronotherm. By heat acceleration, **on** periods of the thermostat may be increased to meet the requirements of the heating plant to assure leveled heat distribution.

The artificial heat is accomplished by utilizing the normal electric current which flows through the instrument. Assuming the thermostat indicator is set at 70°F., then as soon as the surrounding room temperature drops to 70°F., both the blue and white contacts, see Fig. 160, are closed—the white contact having closed at 73°F., assuming a 3-degree differential setting is employed. Referring to Fig. 160 we find that closing the blue and white contacts energizes *D*, closing holding contact *E* and the burner or gas valve circuit is completed through contact *C*. As long as the blue contact is made, heater *F* receives practically no energy, for electrical energy follows the course of least resistance.

As the surrounding temperature rises a fraction of one degree above 70°F., the blue contact will break in response, but the relay will remain energized. The circuit is then through the white contact, holding contact *E*, heater *F*, and the secondary of the transformer.

With the blue contact broken, heater *F* becomes effective and

artificial heat is introduced, which slowly raises the temperature of the thermal element. The artificial heat *accelerates* the normal action of the bimetal, and the white contact is opened faster than usual, causing the burner to stop. Although no further surrounding temperature rise is necessary to stop the burner, yet the construction of the heater is such that, without a further temperature rise, (still assuming a 3-degree differential) it assures continuance of burner operation for approximately 13 minutes after the blue con-

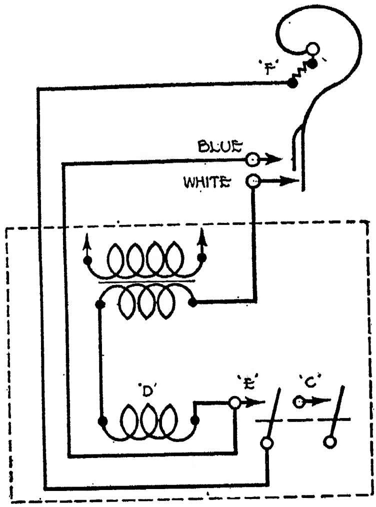


Fig. 160. Schematic Electrical Circuit for the Chronotherm

tact is broken. This may be increased or decreased by adjusting the position of the white contact, but a preliminary room temperature rise will nevertheless be required before any of the artificial heat is applied.

This is a point of major distinction between heat acceleration and heat anticipation, for in heat acceleration, **on** periods may be increased to meet the requirements of the heating plant to assure the heat distribution without reverting to the standard thermostat operation.

Heater *F* is in construction a metal *sponge*. Thus the delivery of heat to the thermal element is delayed a sufficient length of time to assure efficient burner operation, and heat once delivered is retained for sufficient time to prevent undesirable *short cycling*.

Humidity Controls. *Combination Type.* This type has the double function of controlling first, the humidity in an enclosure and second, the temperature, as illustrated by the two control knobs at

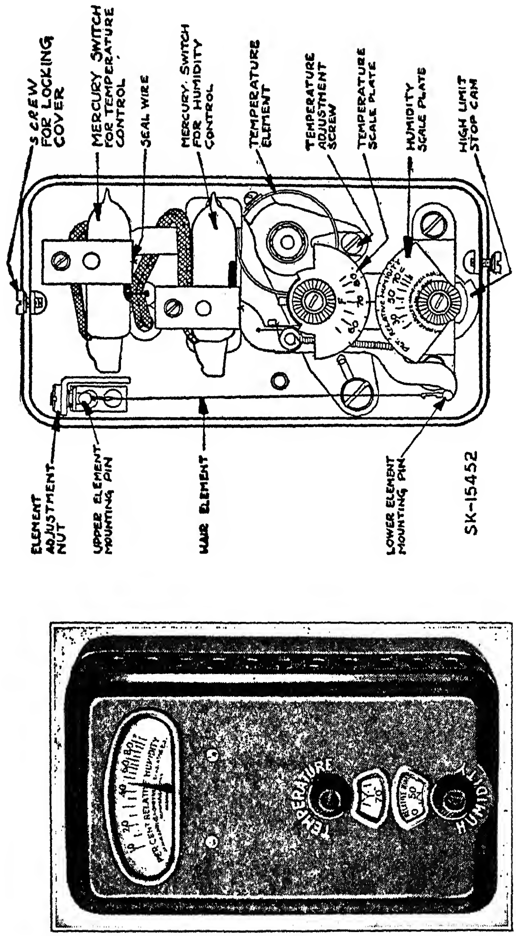


Fig. 161. Combination Humidity-Temperature Control. (This is obsolete but is shown here because many are still in use.)
Courtesy of Minneapolis Honeywell Regulator Company

the left-hand side of Fig. 161 and by the detail drawing at the right-hand side of Fig. 161.

The moisture sensitive element of the humidity control is composed of multiple groups of human hair and provides accuracy and sensitivity in the regulation of relative humidity to a high degree.

The elongation and contraction of this hydroscopic element produced by changes in relative humidity transmits its motion to a mercury switch through a simple lever mechanism. The mercury tube, in turn, provides the electrical switching necessary for operating valves, pumps, dampers, fans, or other means for regulating humidifying or dehumidifying operations. This instrument has a

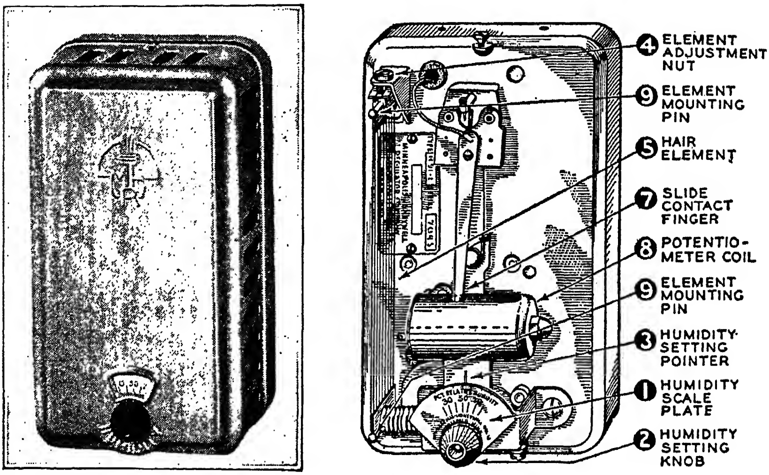


Fig. 162. Modulating Type Humidity Control
Courtesy of Minneapolis Honeywell Regulator Company

bimetal thermostatic element which operates a mercury switch to provide control of motorized valves, dampers, burners, etc., or other means for regulating heat delivery or cooling.

This instrument can be used with all types of air-conditioning systems where a two-position control service is desired.

Modulating Type. This type employs a potentiometer mechanism, Fig. 162, which is actuated by the elongation and contraction of a sensitive hydroscopic element operating through a simple leverage. The slightest change in relative humidity is detected and a sliding arrangement in conjunction with the potentiometer causes a motor operating damper or valve to take up a correspondingly new position. Thus modulation is effected and, as a result, a humidifier or dehumidifier will produce greater or less humidification or dehumidification in exact proportion to the demand. This instrument

is for use in connection with air-conditioning systems where both summer and winter control is desired.

Bulb Temperature Controller. Fig. 163 shows two views of a typical bulb temperature controller so designed that the sensitive bulb can be at any desired distance from the bellows and switching

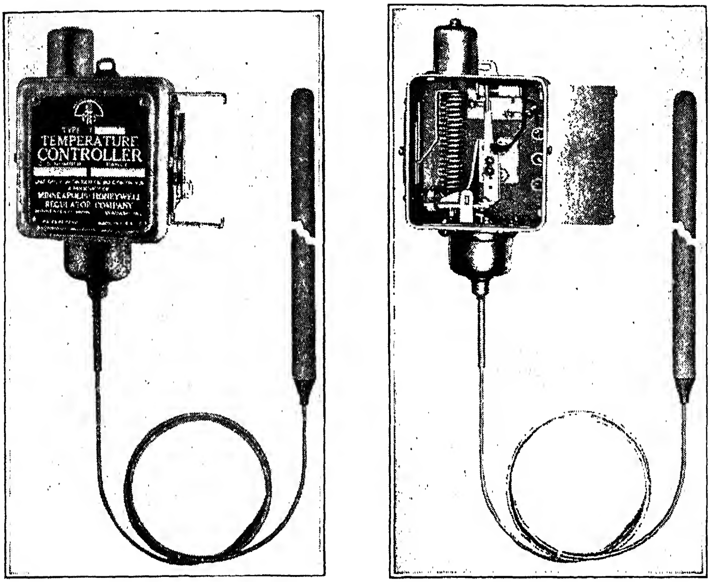


Fig. 163. Bulb Temperature Controller Showing Exterior and Interior Views
Courtesy of Minneapolis Honeywell Regulator Company

mechanism. This type of controller has a wide application, as a means for governing the operation of motorized valves, dampers, heaters, compressors, relays, pumps, or fans applied to air-conditioning systems. It is particularly suitable for duct mounting or other similar mounting where a remote bulb type of instrument presents certain advantages over the wall-mounted thermostat. The operation of the controller is simple and can be used for either the two-position or modulating control of air-conditioning systems.

A change in temperature at the thermostatic bulb or capsule varies the pressure of the gas with which the capsule is charged. This change in pressure is transmitted through the flexible capillary tubing into the actuating bellows which expands or contracts ac-

cordingly. The bellows action is transmitted through a hardened cone shaped pivot to a main operating arm, which, through a lever and linkage arrangement, produces the desired switching action.

Modutrol Motor. Fig. 164 shows a typical modutrol motor of a shaded pole induction type which provides a convenient means of translating demands from controllers operating on functions of

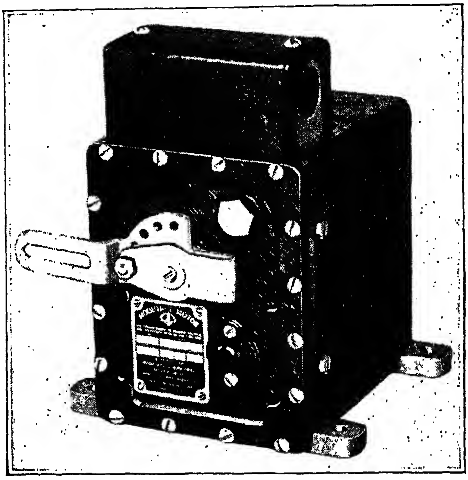


Fig. 164. Modutrol Motor
Courtesy of Minneapolis Honeywell Regulator Company

temperature, pressure, or relative humidity changes into mechanical motion.

The motor is designed especially to serve as the power unit for a number of valve assemblies performing an on and off or two-position service. The field of usefulness for this motor also extends beyond the above important application, however, in that it may be used for the actuation of louver dampers on heating or air-conditioning systems and it can also be used as a motive power for valve applications.

Fig. 165 shows a circuit diagram illustrating the schematic diagram of the modutrol motor. The motor operates as follows:

As a change in temperature (or pressure, or relative humidity, depending upon the function of the controller used) occurs at the controller, contact is made between the common red wire and one

side (either blue or white) of the control circuit which causes rotation of the motor crank arm through one-half revolution. The crank arm will remain in this position until such time as there is a change at the controller opposite to that causing previous operation. When this occurs, contact is made between the red and opposite

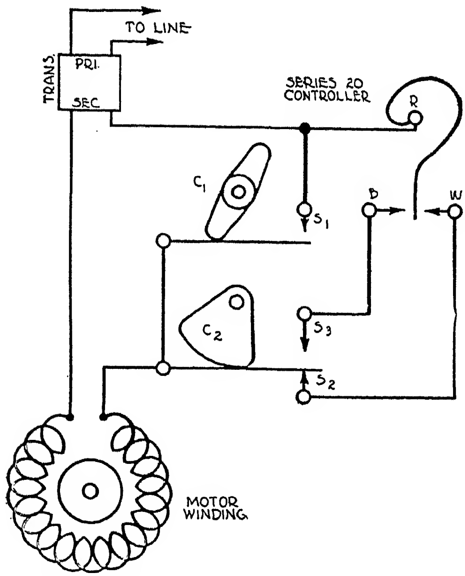


Fig. 165. Circuit Diagram Showing Schematic Diagram of Modutrol Motor

leg of the control circuit causing the motor crank arm to move through the remaining 180° angular degrees to complete a revolution.

Fig. 165 shows schematically the manner in which the transfer switch inside the modutrol motor case provides only momentary starting current through the controller and limits the travel of the motor lever arm to a half revolution for each of the opening and closing operations.

The controller makes contact across the red and blue completing the circuit to the low voltage motor winding through the closed contact at S_3, contact S_1 being open when motor lever arm is at rest. The initial movement of the motor allows cam C_1 in series with the gear reduction train to close and cam C_2, also in series with the train

of gears, to open contact at S_3 and close at S_2. Thus a maintaining circuit is established through S_1 directly to the motor, the blue leg of the control circuit being broken at S_3. The white wire leg of the control circuit is now made at S_2 but broken at W in the controller. As the motor reaches a half revolution, the maintaining circuit is

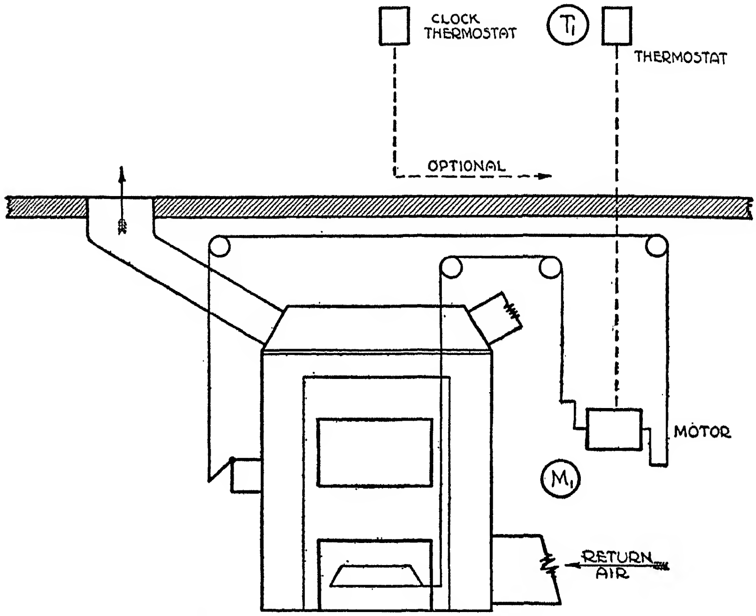


Fig. 166. Control for Coal-Fired Warm-Air Furnace

now broken at S_1 by cam C_1 and the motor stops. When the controller moves from the blue to the white, contact W circuit is completed to the motor through S_2. Again current is only in the controller momentarily as cam C_1 makes contact at S_1 and cam C_2 breaks contact at S_2 and makes at S_3 preparatory to the next controller operation. Thus the motor is successively rotated through half revolutions in a continuous direction as its controller moves successively to bridge first the blue and then the white with the common or red wire of the control circuit.

There are a great many types and kinds of automatic controls some of which have been illustrated and explained in the foregoing pages. Automatic controlling is a science in itself, so no complete

statement of principles or illustration of styles or types can be given or shown in a book of this kind which deals primarily with a kindred subject as the main subject. However, the control instruments which have been illustrated and the central systems which will be illustrated are typical examples and will serve to acquaint the reader with the general method of automatically controlling the heating, ventilating, and air-conditioning systems in sufficient detail to enable him to successfully handle all ordinary control problems.

Operation of Residence Heating and Air-Conditioning Systems. The capacity of a residential heating plant is designed to heat the structure during the most severe weather conditions. When minimum outside temperatures occur, the entire output of the heating plant is in demand and the control problem is largely one of keeping combustion rates within limits of safety. Maximum demand periods are not long nor at frequent intervals so there are other factors of control that predominate. If a good heating system is to keep in close step with outside temperature variations, the controls must be automatic because only automatic control can actually synchronize plant operation with actual demands.

Gravity Warm Air Systems. Fig. 166 illustrates a system of control for a coal-fired warm air furnace that offers exceptionally accurate automatic control from a thermostat. In this system a change in room temperature is measured by thermostat T_1, which causes the damper motor M_1 to reposition the draft and check dampers of the furnace to increase or retard heat as required. An automatically lowered control point can be used, as an optional feature, for reduced night temperatures through the use of a clock type thermostat. This system depends on careful firing by manual means.

The value of the heat actuated thermostat in this, or in any of the control systems shown, is that the thermostat provides, through its cycling action, a definite ceiling for the fire. This prevents the fire from developing too high a head, with correspondingly high stack temperatures, which in turn result in reduced over-all efficiency of the heating plant and overshooting of room temperatures. With the heat actuated thermostat in command of the fire, bonnet temperatures increase or decrease as the outside weather conditions increase or decrease.

By the use of slightly different thermostats, the system shown in Fig. 166 can be so arranged that if the power circuit should fail for any reason the motor will automatically reposition the check and draft so as to check the fire all during the time the electric control circuit is without power. This is a safety feature.

Fig. 167 illustrates a system of control for a coal-fired furnace wherein the room thermostat T_1 maintains definite control of the

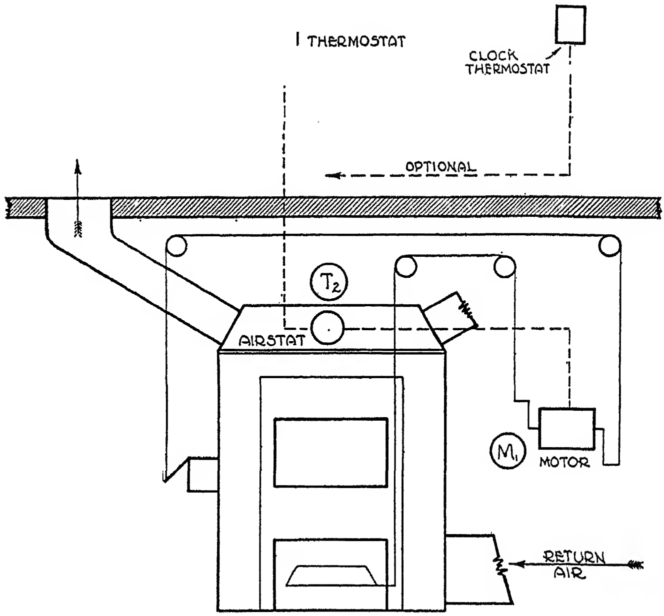


Fig. 167. Control for Coal-Fired Furnace Where the Room Thermostat Works in Conjunction with a Bonnet Control

fire subject to the limitations of a bonnet control thermostat T_2 (or airstat) to prevent excessive bonnet temperatures. This is a combination with the system shown in Fig. 166.

Room thermostat T_1 measures the room temperature and causes damper motor M_1 to position the draft and check dampers in accordance with the temperature demands. Airstat T_2 provides a high limit control to check the fire in case the bonnet rises to an excessive degree. This provides a definite *ceiling* for the fire should the room conditions be such that thermostat T_1 cannot be satisfied without accelerating the fire to a hazardous temperature. During

normal operation, the occasional cycling of the drafts, caused by the action of the heat actuated thermostat T_1, will provide a definite *ceiling* for the fire as described in the control system illustrated in Fig. 168. This system can also be designed to have the safety factor previously mentioned.

Room thermostat T_1 causes the primary controls of the heat generating equipment to start or stop the fire in accordance with the

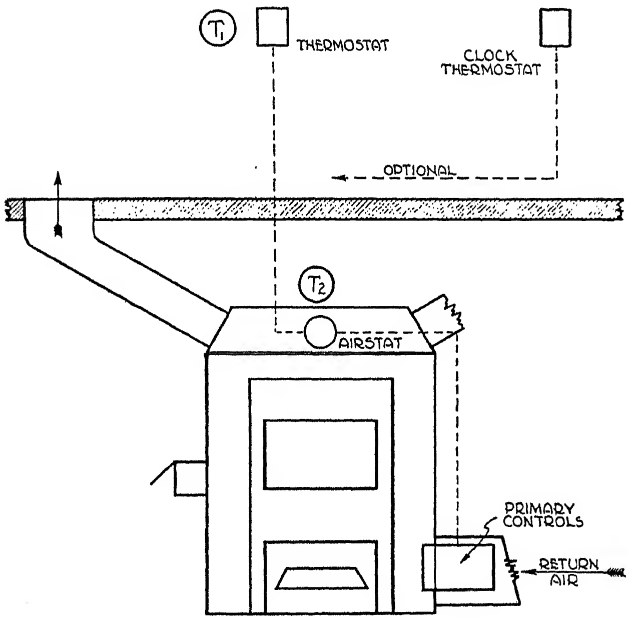


Fig. 168. Control for Automatically-Fired Furnace Where the Room Thermostat Works in Conjunction with a Bonnet Control

changes in room temperature. Airstat T_2 provides a high limit control point that will stop the fire in case the bonnet temperatures become excessive. This provides for a definite limit in the bonnet temperature in case the room conditions are such that thermostat T_1 cannot become satisfied without raising the bonnet to a hazardous temperature.

Fig. 169 illustrates a complete system of control for a coal-fired, forced warm air heating plant. In this system, control of the blower is from bonnet temperature, while control of the fire is from room

temperature. Protection is provided for the heating plant in case of line voltage failure.

The operation of the control system is as follows:

Changes in room temperature are measured by room thermostat T_1, which causes motor M_1 to reposition the draft and check dampers. If the room temperature has been falling, the fire will be accelerated.

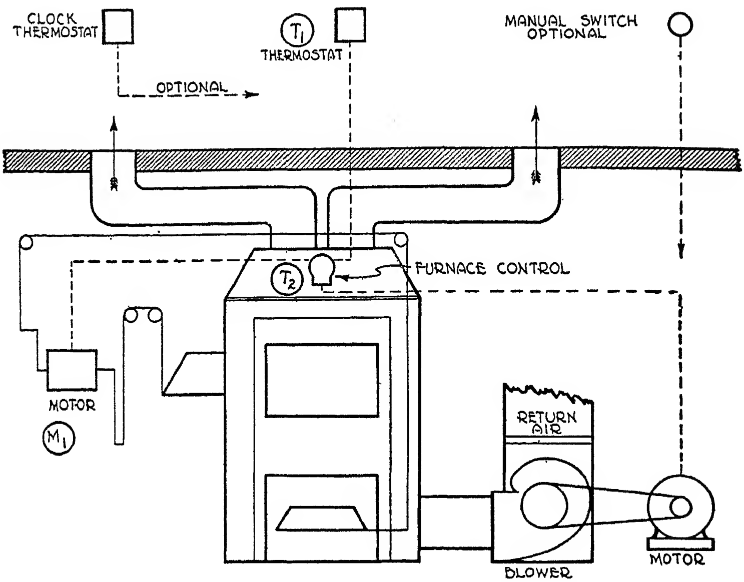


Fig. 169. Control for Coal-Fired Forced Air Furnace

When the bonnet temperature has increased to the setting of the fan control switch in combination furnace control T_2, the blower will start to operate. Should the bonnet temperature continue to rise to a hazardous temperature, the high limit switch in temperature controller T_2 will cause motor M_1 to reposition the dampers and reduce the fire. When thermostat T_1 becomes satisfied, it will cause motor M_1 to reposition the dampers to check the fire, but the blower motor will continue to run until the bonnet temperature has fallen to the point where furnacestat T_2 will cause it to be stopped. Any interruption in line voltage will cause motor M_1 to position the dampers to check the fire.

The control system shown in this figure is very widely used. It is

the most simple type of control system for a forced warm air heating plant. The heat actuated thermostat will maintain fairly constant bonnet temperatures, the level of which will be determined by the outside weather conditions. This will change the temperature of the air at the outlet grilles in keeping with the actual heat losses. On a coal-fired job, particularly with a cast-iron furnace, there may be some tendency to over heat because the blower does not stop, and consequently the heat delivery to the room is not ended when the room thermostat is satisfied. This condition is more pronounced when the heat accelerated thermostat is not used. The heat actuated thermostat will provide much more satisfactory operation.

An interruption of line current will cause the blower of a forced warm air heating system to stop, and since gravity circulation through a forced warm air heating plant equipped with efficient filters is decidedly limited, it is important that the fire be checked immediately to prevent a hazardous condition in the furnace from high temperatures. With this system of control, the fire will automatically be checked in case of current failure due to the action of the spring return mechanism in the damper motor M_1.

Fig. 170 illustrates a system of control for an automatically fired, forced warm air heating plant in which the register air temperatures are varied in response to the actual heat losses of the structure. The volume of the air delivered remains constant. This is a sub-system of control.

The operation of the control system is as follows: A change in room temperature is measured by room thermostat T_1, which repositions modutrol motor M_1 operating the mixing dampers in the tempered air chamber. Motor M_1 changes the relative position of the mixing damper, creating a change in the temperature of the tempered air sufficient to meet the requirements of room thermostat T_1. Ductstat T_2 operates as a low limit to prevent air from being circulated at temperatures which would cause unsatisfactory conditions.

Heating systems of this type depend upon a constant supply of warm air in the warm air plenum chamber. The damper motor may be equipped with an auxiliary switch if desired, which will stop the burner when the damper to the warm air plenum chamber is in the fully closed position. Proper insulation of the heating plant, espe-

cially of the warm air plenum chamber, is very essential on a heating system of this kind.

Zone Control System. The inherent weakness of a single thermostat temperature control system, for all but the very smallest type of residence, lies in the fact that all parts of the structure do not lose heat at a uniform rate. The heat losses of the various sections of the

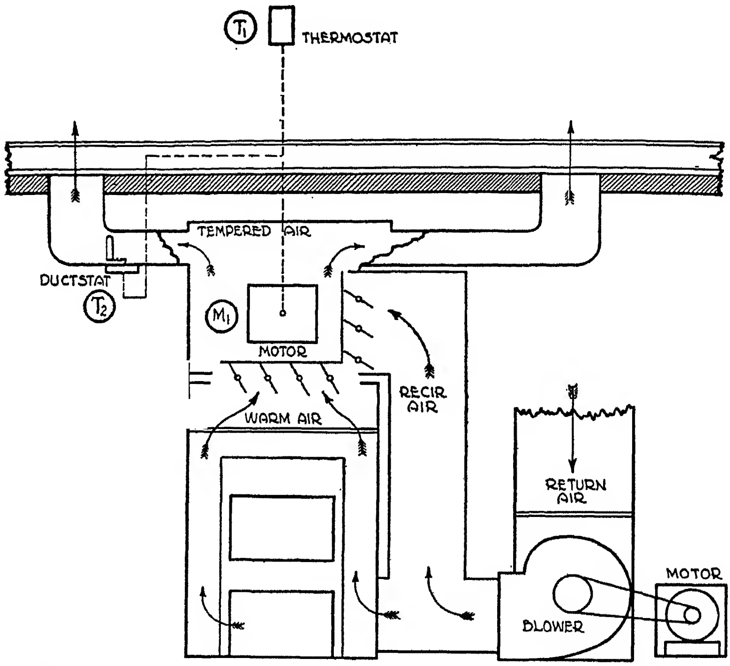


Fig. 170. Controls for Automatically-Fired Forced Air Furnace in Which the Register Temperatures Are Varied According to Heat Losses

residence, or even of the various rooms, change from day to day, if not even from hour to hour, because of changes of occupancy and the shifting of the several weather variables. Changes of occupancy involve the number of occupants as well as the type of occupancy. The weather variables include the effect of the sun, the wind direction, the wind velocity and the dry-bulb temperature.

If there exists a definite set of weather variables and a definite type of occupancy, it would be possible to balance the heat deliveries through the ducts of a forced warm air heating system by adjusting

the duct dampers so that absolutely uniform room temperatures could be maintained throughout the building. It is obvious, however, that occupancy does vary, and that the weather conditions vary through a wide range. The rooms of a house exposed to the bright afternoon sun require less heat than the same rooms will require on

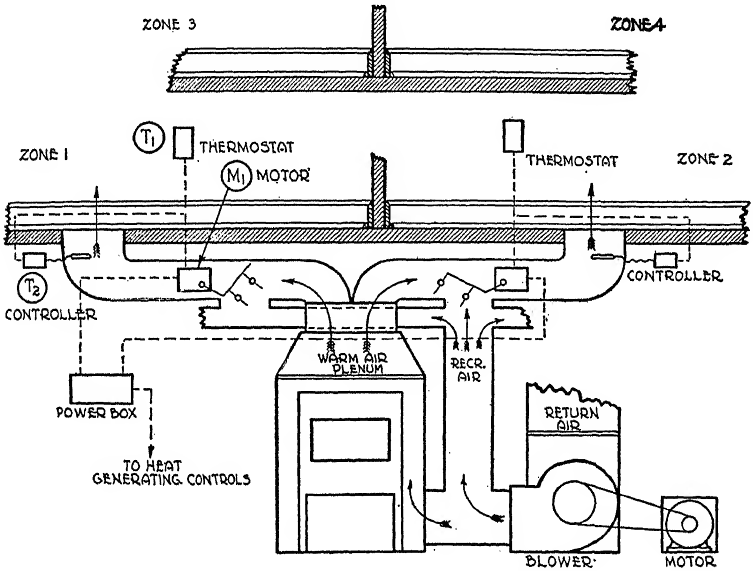


Fig. 171. Zone Control Using Modulating Equipment

a day when there is no sun, and likewise the rooms of a building on the windward side require far more heat than those rooms on the side of the building protected from the wind. Occupancy varies with the number of people in the room as well as with the type of activity in which they are engaged. The only way to compensate for these changing conditions is to vary the heat input into the individual rooms, or into a group of rooms from which the heat loss characteristics are approximately the same.

Fig. 171 illustrates a system of zone control using modulating equipment, in which the air delivered at the register remains constant but the air temperature at the register varies in accordance with the heat demands of the room. Since the air is in constant circulation at all times, stratification is lessened, and even temperatures are more probable in all parts of the zone.

The operation of the system is as follows: The temperature of the zone, measured by thermostat T_1, determines the position of motors M_1, and consequently the position of the dampers in heated air and recirculated air supply. As the room temperature falls, thermostat T_1 will cause motor M_1 to reposition the dampers until the mixture of heated air and recirculated air is of proper temperature to off-set the heat losses of the zone. Conversely, as the temperature rises, the dampers will be positioned by motor M_1 until the air mixture in the duct contains less heated air and a sufficiently greater amount of recirculated air to maintain the proper room temperature. Controller T_2 is located in the zone duct between the mixing dampers and the discharge grille. Its purpose is to prevent air at too low a temperature from being forced into the zone. The control setting of controller T_2 should be determined by the design of the heating system. Air from baseboard registers must be of a higher temperature than air coming from high wall-mounted registers if drafts are to be avoided. The setting will also be influenced to a considerable degree by the register velocities for which the system is designed.

The auxiliary switch on motor M_1 is arranged so that it will be open when the damper is in the position where all recirculated air is used, and the damper to the heated air plenum chamber is entirely closed. The auxiliary switches, wired through the power box, are provided to control the heat generating equipment. When all the zone thermostats have become satisfied, and all the M_1 motors have moved into the position where all recirculated air is used, the control circuit to the heat generating equipment is open, and heat is no longer generated. The fan continues to run until stopped manually. When one of the zone thermostats again calls for heat, the M_1 motor associated with it, will move to open its damper to the heated air plenum chamber, and by so doing will close the contact on its auxiliary switch. The closing of this circuit will cause the heat generating equipment to start and be placed under the command of a limit control in the heated air plenum chamber.

With this system of zone control, the blower operates continuously throughout the heating season. It can be combined with other sub-systems of control in which the bonnet temperature is maintained at a minimum temperature continually, if desired.

Air-Conditioning System. Fig. 172 illustrates a complete system for the control of an all-year residential air-conditioning system using forced, warm air heat with a warm air furnace as the primary heat source. A manual change-over is provided for changing from summer to winter operation.

The operation of the control equipment is as follows:

When the summer-winter switch S_1 is in the winter position, the solenoid valve on the high side liquid line of the compressor is closed.

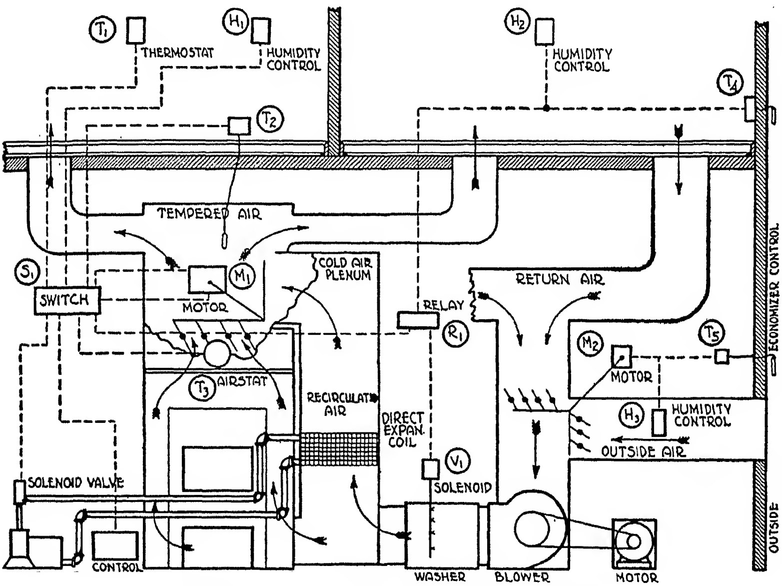


Fig. 172. Controls for All-Year Air-Conditioning System

Temperature controller T_3, located in the warm air plenum chamber of the furnace, will control the oil burner to maintain a definite bonnet temperature in the warm air plenum chamber.

Thermostat T_1 will measure the temperature in the room and will modulate motor M_1 which in turn will position the mixing dampers to regulate the relative quantities of warm air and recirculated air to maintain a definite room temperature.

Temperature controller T_3 will act as a low limit control, and will take command of motor M_1 whenever necessary to prevent the tempered air plenum chamber from falling below a minimum temperature.

Humidity controller H_2 will measure the relative humidity in the room and will operate relay R_1 subject to the compensating effect of outside compensator T_4. Relay R_1, responding to the action of humidity control H_2, will open solenoid water valve V_1, admitting water to the sprays and providing additional humidity when required. Water valve V_1 cannot open if the blower is not in operation.

Temperature controller T_5 will measure the outside air temperature and will modulate motor M_2 to position the mixing dampers in the return air and outside air ducts to admit all outside air when the outside temperature is high enough to reduce the load on the heating plant. The blower operation will be continuous and is controlled by manual switch.

When switch S_1 is in the summer position, the oil burner control, through temperature controller T_3, will be inoperative.

Thermostat T_1 will measure the room temperature and modulate motor M_1 which will position the mixing dampers to regulate the relative quantities of cooled air and recirculated air to maintain a definite room temperature. The dual control switch of motor M_1 will open the solenoid valve on the high side of the liquid line from the compressor when the mixing dampers to the cold air plenum chamber start to open from the fully closed position.

Temperature controller T_2 will act as a low limit control and will assume command of motor M_1 to reposition the mixing dampers when it becomes necessary to prevent the temperature in the tempered air plenum chamber from falling below a minimum point.

Humidity control H_1 measures the relative humidity in the room, and will assume command of motor M_1 and the mixing dampers whenever the relative humidity rises above a predetermined point.

Temperature controller T_5 will measure the outside temperature and will modulate motor M_2 to admit outside air whenever the outside temperature is low enough to reduce the cooling load on the cooling equipment. Humidity control H_3 will take command of motor M_2 and close the outside air damper if the outside humidity is excessively high.

The control circuit to relay R_1 will be in the open position so it cannot operate solenoid valve V_1 and admit water to the sprays when the cooling equipment is in operation.

CHAPTER XIII

TYPICAL EXAMPLE

*__Example.__ In this example it is assumed that the residence shown in Figs. 173 to 176, inclusive is perhaps about 18 or 20 years old and that it was damaged by fire and water to the extent that most of the plaster, roof, and attic flooring has to be replaced. It is further assumed that the owner is desirous of not only repairing the structure but also feels that a new heating system is necessary to replace a worn out gravity system that had been in use. The owner has decided to install a complete air-conditioning system of the forced air type for heating, cooling, humidifying, dehumidifying, and cleaning the air, using gas as fuel.

It is required to make up a complete recommendation and plan for this type of air-conditioning system.

Before any thought can be given to the actual design of apparatus, the residence must be carefully inspected and its construction features noted in detail. Full information as to dimensions, windows, doors, etc., can be assumed here the same as shown in Figs. 173 to 176, inclusive. The following added information is obtained.

Walls: All outside walls and inside partitions are of frame with 2×4's as studs. The balloon type framing was used. The outside walls are constructed of wood sheathing, paper, and siding, on the outside, and wood lath and plaster on the inside. All partitions are wood lath and plaster on both sides of the studs.

Floors: The attic floor, constructed of rough flooring on 2×8 joists, was so badly burned as to require replacing together with many joists. The second floor has rough and finish flooring over the 2×10 joists. The first floor is the same as the second floor.

Ceilings: The second floor ceiling was almost entirely destroyed. The first floor ceiling was so badly soaked with water as to require replacing.

Foundation: 12-inch concrete entirely below grade.

Basement: The basement floor is 4 inches thick and of concrete. There is no ceiling other than bare 2×10's.

Roof: The roof was badly damaged and requires complete rebuilding.

Windows and Doors: Single double hung windows of average quality. The doors are 1⅝-inch wood.

Playroom: Concrete floor. Side walls are of wood lath, and plaster furred out from concrete. Ceiling is of wood lath, and plaster. Inside partition is of wood lath and plaster on studs.

Experience has taught that a house of this type offers very little thermal resistance because all materials have a high transmission rate.

Note: This can be checked by use of Tables 2 to 15, inclusive.

***This example aims to present the job of calculating heating and cooling requirements as explained in the second and third paragraph of the special note on p. 174.**

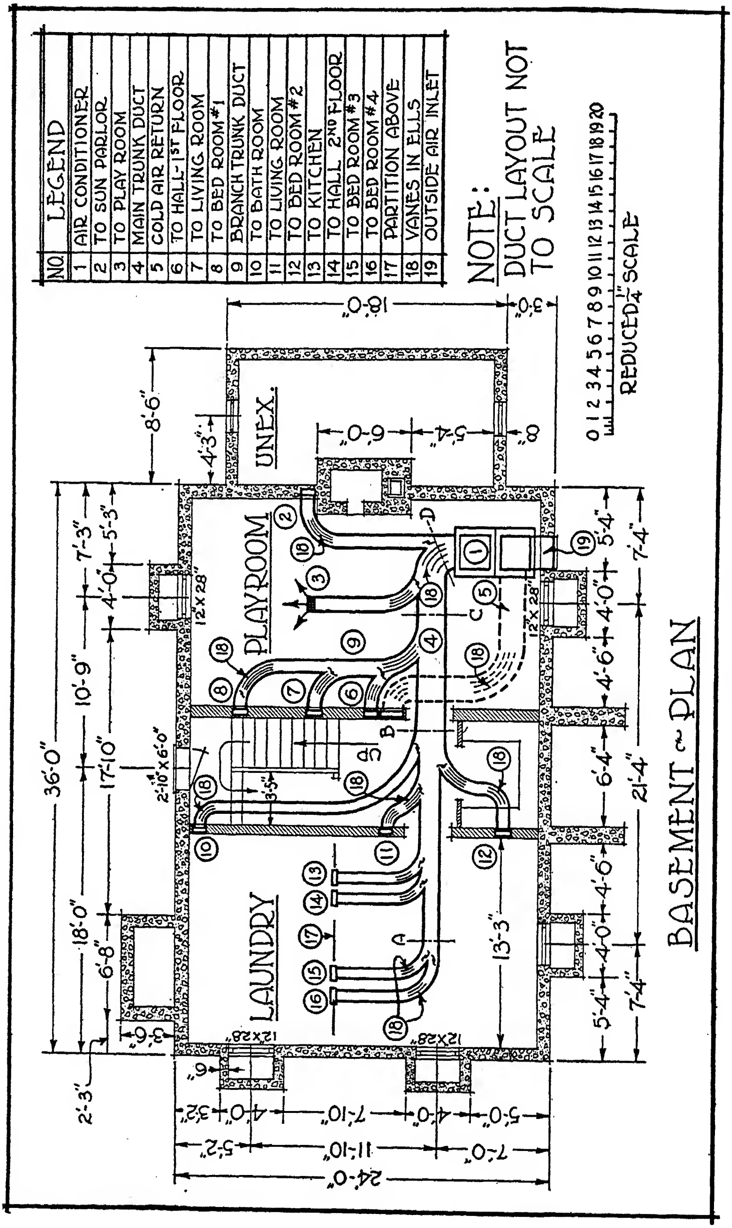


Fig. 173. Basement Plan

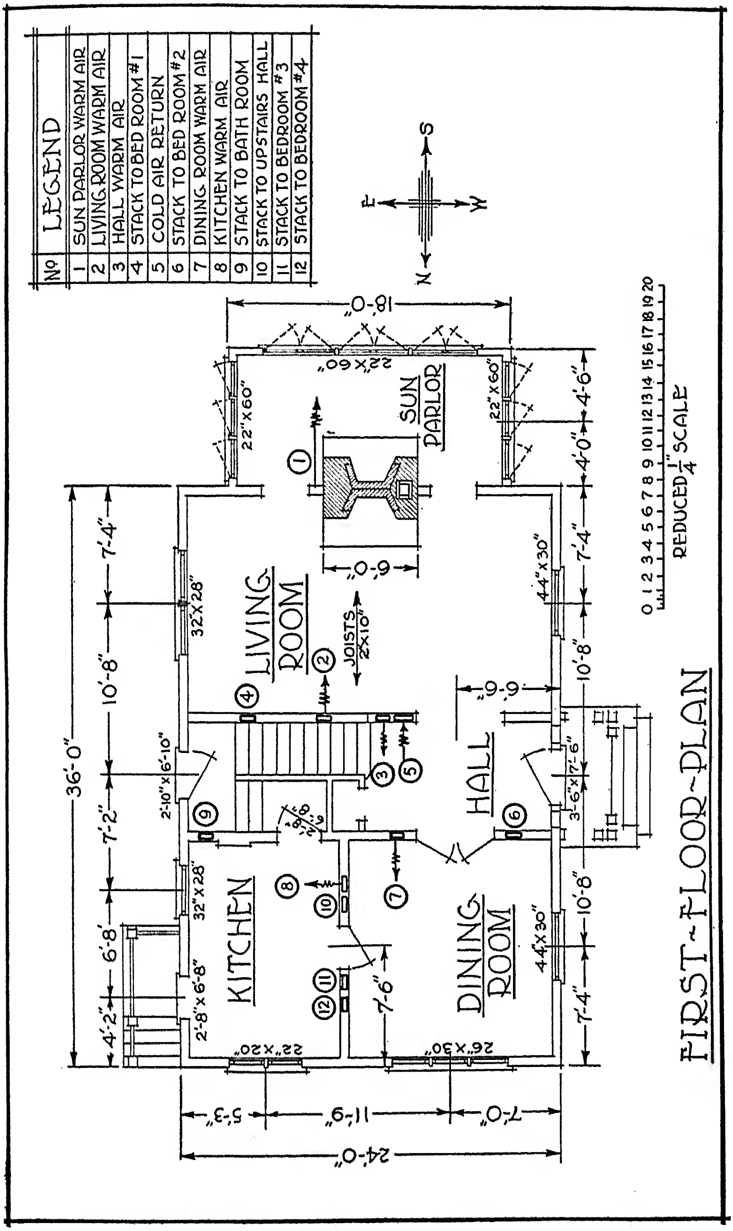


Fig. 174. First Floor Plan

Knowing that an air-conditioning system would prove too costly, both as to first cost and maintenance in this house as it originally stood, it is evident that some form of insulation should be applied during the repair operations.

There are several types of insulation, as shown and explained in Vol. II, almost all of which could be applied in this case. Knowing that all lathing and plaster must be replaced, it seems most logical to use insulation instead of new laths. This would serve the double function of providing thermal resistance and, because such rigid insulating plaster backing may be obtained in large sheets, it would give the weakened framework a general stiffening and strengthening effect. This last item is important in a structure that has been weakened by fire. As no other additional insulation is thought economical, the plaster backing is specified ½-inch thick and applied directly to the studs, as shown in Table 14, Vol. II. This shows types from which Insulite is selected as a typical kind. This insulation is recommended for all walls and ceilings. Ordinarily the first floor ceiling could be lathed with wood or metal laths but the expense for insulation is very little more and provides the first floor with insulation from the second floor in case only one floor at a time is heated or cooled.

The roof has to be rebuilt and as it receives great heat from the sun, it is recommended that ⅞-inch Insulite sheathing be substituted for roof boards and that the shingles be applied to 1×2-inch furring strips nailed through the insulation and into the rafters.

Weatherstripping is recommended for all windows. It is also recommended that awnings be used for all south and west windows during the summer months. The residence faces south. Assume Chicago as the location.

Knowing all the foregoing specifications, the heating and cooling loads may be calculated as follows.

Heating Load. The Technical Code recommendations are used in calculating the heat losses in terms of H, for each room and H_b for the entire residence.

Note: The Technical and Gravity Codes, together with the B.t.u. method, aim at the calculation of heat losses. Their respective procedures vary somewhat and their results may even vary but in general the results are close enough so no trouble would be encountered. In like manner, Vol. II, Chapter V shows a different formula for calculating heat loss than is given in the Technical Code. Here, too, the results by using either formula are practically the same.

The Technical Code and all practical methods require that net wall areas, glass areas, roof areas, ceiling areas, and floor areas be known. Therefore this information is calculated first. Table 92 is used to record the information as it is found or calculated.

The glass, net wall, etc., areas are first calculated for the living room. The length is 24′0″ minus the thickness of two walls or 24′0″−1′0″=23′0″. The width is 14′6″ minus the thickness of one wall plus half of another or 14′6″−0′9″=12′9″.

Note: It must be remembered that dimensions from *inside* of walls are used.

The whole length of the living room walls is not considered because most of it is either bounded by an inside partition (no temperature difference) or by the sun parlor. Thus only two short lengths must be considered beyond either end of the sun parlor. By scaling, each length is assumed as 2′6″ or 5′0″ together. The ends are 12′9″+12′9″ or 25′6″. Then 25′6″+5′0″=30′6″ total length of exposed wall for living room. Fig. 176 shows the ceiling height to be 10′0″ for the first floor. So 30′6″×10′0″=305 square feet gross area.

The window or glass area is next calculated. At the rear end there are two windows each 32″×28″* or 2′8″ × 4′8″. The area of these two windows is 2(2′8″×4′8″) = 2×11 = 22 square feet approximately. At the front end there is one window 44″×60″* or 3′8″×5′0″. Its area is 3′8″×5′0″ = 19 square feet approximately. Then 22+19 = 41 square feet glass area. The *net* wall area is then 305−27 = 264 square feet. This information relative to the net wall and glass area is found in Table 92. There are no outside doors in the living room so that place in Table 92 is left blank. The living room ceiling is not considered because the second floor is also heated to 70°F. The floor is not considered because the playroom is heated to 70°F.

The sun parlor (room to right of living room) is 18′0″−1′0″ or 17′0″ long. It is 8′6″−0′6″ or 8′0″ wide. The exposed wall is therefore 17′0″+16′0″ = 33′0″ long. Then 33′0″×10′0″ = 330 square feet gross area. There are 12 windows each of which is 22″×60″ or 1′10″×5′0″ or a combined area of approximately 108 square feet. The *net* wall area is then 330−108 = 222 square feet. The sun parlor floor is above a cold space. Therefore its area, 136 square feet, must be considered and is found in Table 92. The ceiling is also considered as being exposed and has the same area as the floor.

The first floor hall has a short exposed wall. Its length is 7′0″−1′0″ or 6′0″. Its gross area is 6′0″×10′0″ or 60 square feet. The door dimensions are 3′6″×7′6″ or an area of approximately 25 square feet. The *net* wall area is then 60−25 = 35 square feet. The landing in the stairwell has an exposed wall which must be considered. Its exposed wall extends to the second floor ceiling. If we assume the landing half way between the first floor ceiling and floor, and that the second floor ceiling is 9′0″ (See Fig. 176), then the height of this wall is 9′0″+5′0″ or 14′0″. The exposed wall has a width of 6′0″. Its area is then 14′0″×6′0″ or 84 square feet, gross area. The door in this area is 2′10″×6′10″ or approximately 18 square feet. The *net* area is 84−18 = 66 square feet. Then 35+66 = 101 square feet *net* wall area for hall. The two doors equal 25+18 = 43 square feet. The ceiling cannot be considered, except as shown for the upstairs hall. The floor area is 6′0″×13′0″ (scaling) or 78 square feet.

In like manner all other first floor rooms are considered.

Note 1. Generally a certain amount of estimating is done in making such calculations so that it is not probable that any two engineers would obtain exactly the same figures as shown in Table 92.

Note 2. The lineal feet of crack for each window or door has been calculated in the same manner as explained for the factory building in Vol. II. Where there is some framing between double windows, as in the case of the two windows in the rear of the living room, each window must be figured separately. The windows in the sun parlor are double but only one crack is figured between them. Crack lengths may be figured somewhat approximate. The Code specification in Section 2 is used to determine how much crackage to use.

In the living room there are three exposed walls. There the wall having the greatest length of crack is considered. This is the rear wall containing two windows. Therefore the crackage for these two windows is calculated approximately and is found in Table 92.

For the first floor hall, which was considered as having two exposed walls, the front wall has the large door (doors are considered the same as windows), so its crackage is found in Table 92.

In the sun parlor the windows are casement style and therefore have no

*Length is twice this amount.

*Table 92. Calculation Sheets Showing Basic Figures Used in Estimating Heat Losses of House in Figs. 173 to 176 Inclusive

Rooms	Net Exposed or Other Area Sq. Ft.	U Values	Temp. Diff. °F.	Lineal Feet of Crack ‖	Loss in B.t.u. Per Hour	Totals
Living Room						
Wall	264	.19	80	..	4,011	
Floor	...	....	..	..	.....	
Ceiling	...	....	..	..	.....	
Glass	41	1.13	80	..	3,706	
Door	...	....	..	..	.....	
†Door Crack	...	....	..	..	.....	
†Window Crack	...	....	80	35	1,189	8,906
Hall First Floor						
Wall	101	.19	80	..	1,535	
Floor	78	.34	30	..	796	
Ceiling	...	....	..	..	.....	
Glass	...	....	..	..	.....	
Door	43	1.13	80	..	3,887	
†Door Crack	...	....	80	25	850	
†Window Crack	...	....	..	..	.....	7,068
Dining Room						
Wall	203	.19	80	..	3,086	
Floor	178	.34	30	..	1,816	
Ceiling	...	....	..	..	.....	
Glass	63	1.13	80	..	5,695	
Door	...	....	..	..	.....	
†Door Crack	...	....	..	..	.....	
†Window Crack	...	....	80	49	1,665	12,262
Sun Parlor						
Wall	222	.19	80	..	3,374	
Floor	136	.34	80	..	3,699	
Ceiling	136	.62	80	..	6,746	
Glass	108	1.13	80	..	9,763	
Door	...	....	..	..	.....	
†Door Crack	...	....	..	..	.....	
†Window Crack	...	....	80	69	2,335	25,917
Kitchen						
Wall	199	.19	80	..	3,025	
Floor	129	.34	80	..	3,509	
Ceiling	...	....	..	..	.....	
Glass	20	1.13	80	..	1,808	
Door	12	1.13	80	..	.....	
†Door Crack	...	....	80	18	612	
†Window Crack	...	....	80	18	612	9,566
Bedroom No. 1						
Wall	210	.19	80	..	3,192	
Floor	...	....	..	..	.....	
Ceiling	168	.21	54	..	1,905	
Glass	24	1.13	80	..	.....	
Door	...	....	..	..	.....	
†Door Crack	...	....	..	..	.....	
†Window Crack	...	....	80	17	578	5,675
Bedroom No. 2						
Wall	229	.19	80	..	3,481	
Floor	...	....	..	..	.....	
Ceiling	190	.21	54	..	2,155	
Glass	25	1.13	80	..	2,260	
Door	...	....	..	..	.....	
†Door Crack	...	....	..	..	.....	
†Window Crack	...	....	80	19	646	8,542
Bedroom No. 3						
Wall	203	.19	80	..	3,086	
Floor	...	....	..	..	.....	
Ceiling	159	.21	54	..	1,803	
Glass	26	1.13	80	..	2,350	
Door	...	....	..	..	.....	
†Door Crack	...	....	..	..	.....	
†Window Crack	...	....	80	19	646	7,885

*Table 92. Continued

Rooms	Net Exposed or Other Area Sq. Ft.	U Values	Temp. Diff. °F.	Lineal Feet of Crack ‖	Loss in B.t.u. Per Hour	Totals
Bedroom No. 4						
Wall	157	.19	80	..	2,386	
Floor	...	....	..	..	.....	
Ceiling	86	.21	54	..	975	
Glass	18	1.13	80	..	1,627	
Door	...	....	..	..	.....	
†Door Crack	...	....	..	..	.....	
†Window Crack	...	....	80	16	640	5,628
Bathroom						
Wall	45	.19	80	..	684	
Floor	...	....	..	..	.....	
Ceiling	49	.21	54	..	465	
Glass	4	1.13	80	..	362	
Door	...	....	..	..	.....	
†Door Crack	...	....	..	..	.....	
†Window Crack	...	....	80	8	272	1,783
Hall Second Floor						
Wall	...	....	..	..	.....	
Floor	...	....	..	..	.....	
Ceiling	103	.21	80	..	1,730	
Glass	...	....	..	..	.....	
Door	...	....	..	..	.....	
†Door Crack	...	....	..	..	.....	
†Window Crack	...	..	..	..	.....	1,730
Playroom						
Wall	335	.34	45#	..	5,116	
Floor	297	1.07	45	..	14,300	
Ceiling	...	....	..	..	.....	
Glass	5	1.13	80	..	452	
Door	16	1.13	80	..	1,446	
†Door Crack	...	....	30	9	115	
†Window Crack	...	....	80	18	612	
Partition	150	.62	30	..	2,790	24,831
					Grand Total	119,793

*Figures are approximate.
†Infiltration is figured by using the figures shown on this line plus Formula (37). Infiltration is figured for both windows and doors.
‖See Note 2, p. 291.
#The basement (playroom) walls are plastered so no temperature increase is made as suggested in the Technical Code (Item a, Section 2, Article 6).

meeting rails. The windows are in pairs with no framing between them so for each pair three vertical cracks and two horizontal cracks are considered.

For the dining room only the three windows on the left-hand wall are considered. All other crackage is calculated in the same manner.

Bedroom No. 1 has a length of 14′6″−0′9″ (wall thicknesses) or 13′9″. The width is 12′3″−0′9″ or 11′6″. There are two exposed walls as per the dimensions just calculated. The chimney is disregarded and the full length of wall is assumed. Then 13′9″+12′3″=26′0″. Then 26′0″×9′0″=234 square feet gross area of exposed walls. The glass area is approximately 24 square feet for the two windows. The *net* wall area is then 234−24=210 square feet. The floor area need not be considered because the space below is also 70°F. The ceiling must be considered. Its area is 13′9″×12′3″ or 168 square feet. The closet is considered as part of the room.

For the second floor hall only the ceiling need be considered because its exposed wall was included with the first floor hall. The ceiling area is 6′6″×11′6″ or 75 square feet. To this is added 28 square feet to take care of that portion of the hall ceiling adjacent to the bathroom. Thus 75+28=103 square feet.

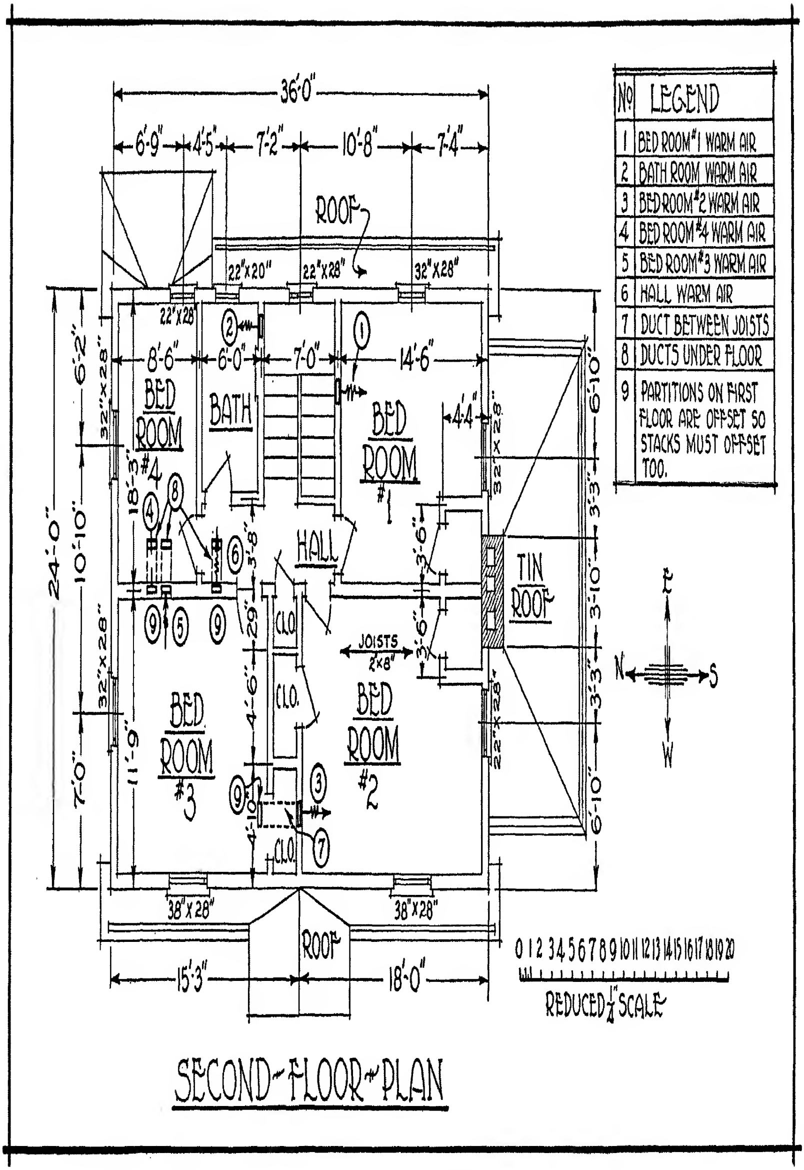


Fig. 175. Second Floor Plan

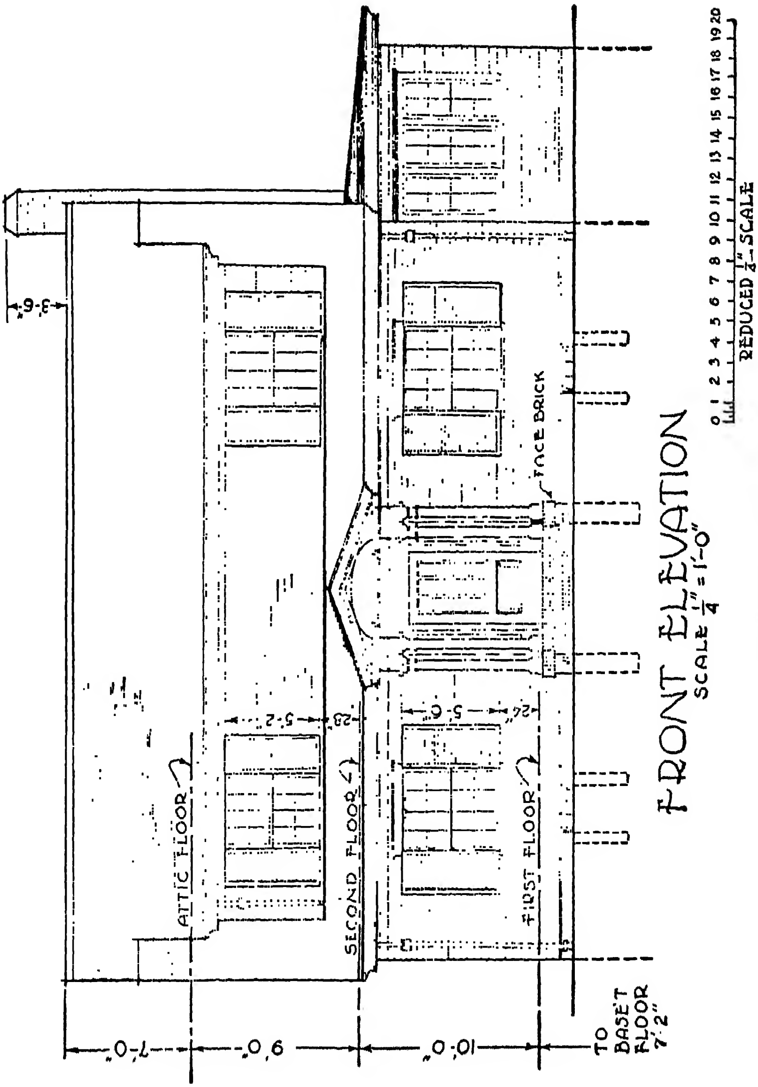


Fig. 176. Front Elevation

All other rooms are calculated in the same manner, except that in bedrooms Nos. 2 and 3 the closets between them may be disregarded because they are so small.

The playroom is 24′0″−1′8″ (thickness of two foundation walls) or 22′4″ long. It is 13′4″ wide. The gross area is (22′4″+13′4″+13′4″) 7′6″ (assume basement depth) or 340 square feet. The glass area is approximately 5 square feet. Then the net wall area is 340−5=335 square feet. The ceiling need not be considered. The floor area is 22′4″×13′4″ or 297 square feet. The crackage is approximately 9 lineal feet. The basement window is divided into three parts so that there are 4 vertical cracks each 1 foot long and 2 horizontal cracks each 2′4″ long. According to Section 2 of the Code only one window need be included for crackage.

Note: The chimney is disregarded and it is assumed the wall is continuous.

The door is 2′8″×6′8″ or an area of approximately 16 square feet. Its crack is approximately 18 feet long.

Note: The Code does not specify a condition where windows or doors are on three sides of a room so only one window and the door are considered.

The temperature differences are next considered. Table 55, Chapter VI shows that the design temperature to use for Chicago is −10°F. The inside temperature is 70°F. Therefore the temperature difference is 80°F. For the first floor hall, the 80° applies for the wall and door area but not for the floor. The basement hall and laundry can be assumed at a temperature of 40°F. Therefore the temperature difference for the first floor hall floor is 70°−40° or 30°. For the dining room the 80° applies in all cases, except for the floor where 30° is used. The sun parlor has an open space under it which becomes as cold as outdoor air. This is because there are louvers at both ends of this space. Therefore the 80° temperature difference applies to all sun parlor considerations. The ceiling is also 80° difference because the roof is very close to the ceiling and no exceptions are assumed. All other first floor temperature differences are determined in like manner.

For the second floor rooms the conditions are somewhat different. The walls have the 80° temperature difference but the ceiling is governed by Table 51. In bedroom No. 1, as in all other parts of the second floor, the ceiling is adjacent to an unheated attic. The first consideration is therefore to find the temperature of the air in the attic. This can be assumed or taken from Table 51 if the value of U_c is approximately the same as any of the V_c values given in Table 51. The symbol U_c means rate of heat transmission through the ceiling, or, in other words, simply the U value of the ceiling construction. From the information obtained from the examination of the residence and the recommendations, it was determined that the ceiling is to be 1-inch Insulite rigid plaster backing and plaster. The attic floor is to be rough flooring only. Table 8, Vol. II, shows that this construction has a U value of .21. This coincides with the .2 in Table 51. Then under the −10°F. design temperature the value of t_a is seen to be 16°F. Therefore the temperature difference between the attic air and bedroom air is 70°−16°=54°F. This same temperature difference applies to all other ceiling areas on the second floor.

For the basement wall the Code, in Section 2, allows a ground temperature of 25° in zero weather. This is considered close enough for use here. The inside

temperature of the playroom is 70°. Therefore the temperature difference for the wall is 70° − 25° or 45°F. The temperature difference for the concrete floor must be estimated. It can be assumed that the air at the floor level is 65°F. The Code allows a ground temperature of 20° below room temperature or 50°. Then 65° − 20° = 45°F. The basement hall was assumed as 40°F. Thus, for the playroom door the temperature difference is 70° − 40° or 30°F.

Next, the U values must be determined. The U value for the frame walls is found in Table 5, Vol. II, and is .19. For the first floor construction the U value is found in Table 8, Vol. II, to be .34. The second floor ceiling U value is .21, as previously explained. The U value for single glass is given in Vol. II, Table 13, as 1.13. The Code assumes doors the same as windows because of their thin construction. Their U values are therefore assumed as 1.13. The sun parlor ceiling is assumed as wood lath and plaster on joists. The U value of .62 is found in Vol. II, Table 8. The concrete walls around the playroom are 10-inch concrete and the U value of .34 is found in Vol. II, Table 3. The partition wall in the playroom is plastered on one side and has bare studs on the other side. Table 6, Vol. II, shows a U value of .62. Table 10, Vol. II, shows a U value of 1.07 for the concrete floor.

Note: All U values have been determined from Vol. II, Tables 2 to 13, inclusive. In cases where U values cannot be determined from the tables, they must be calculated as illustrated in Section 8 of the Code.

The calculation of heat loss per hour is the final step in determining the heating load. The losses for the living room are considered first. Formulas (29) to (40) in Article 3 of the Code are used. Formula (29) is used in finding the loss through glass.

The Code assumes glass and doors one and the same. From Table 92 the glass area is taken as 41 square feet. Writing Formula (29) and substituting gives

$$\begin{aligned}\text{B.t.u. per hour} &= U_g A_g (t_i - t_o)\\ &= 1.13 \times 41 \times 80\\ &= 3{,}706\end{aligned}$$

The meanings of all formula letters are given in Section 2 of Article 1. The 3,706 is found in Table 92.

The loss through exposed walls is calculated by using Formula (30).

$$\begin{aligned}\text{B.t.u. per hour} &= U_w A_w (t_i - t_o)\\ &= .19 \times 264 \times 80\\ &= 4{,}011\end{aligned}$$

The infiltration loss is calculated by Formula (37).

$$\begin{aligned}\text{B.t.u. per hour} &= .018CL(t_i - t_o)\\ &= .018 \times 23.6 \times 35 \times 80\\ &= 1{,}189\end{aligned}$$

The 23.6 is obtained from Vol. II, Table 19. The 1,189 is found in Table 92 in the space opposite to the infiltration space.

Losses for all other rooms are figured in the same manner.

It will be noted that the temperature difference is not the same in all cases as previously explained.

The total heat loss for the heated areas of the house amounts to 119,793 B.t.u. per hour.

Miscellaneous Considerations. The usual operating practice for heating with a mechanical warm air system is to use 100 per cent recirculation as-

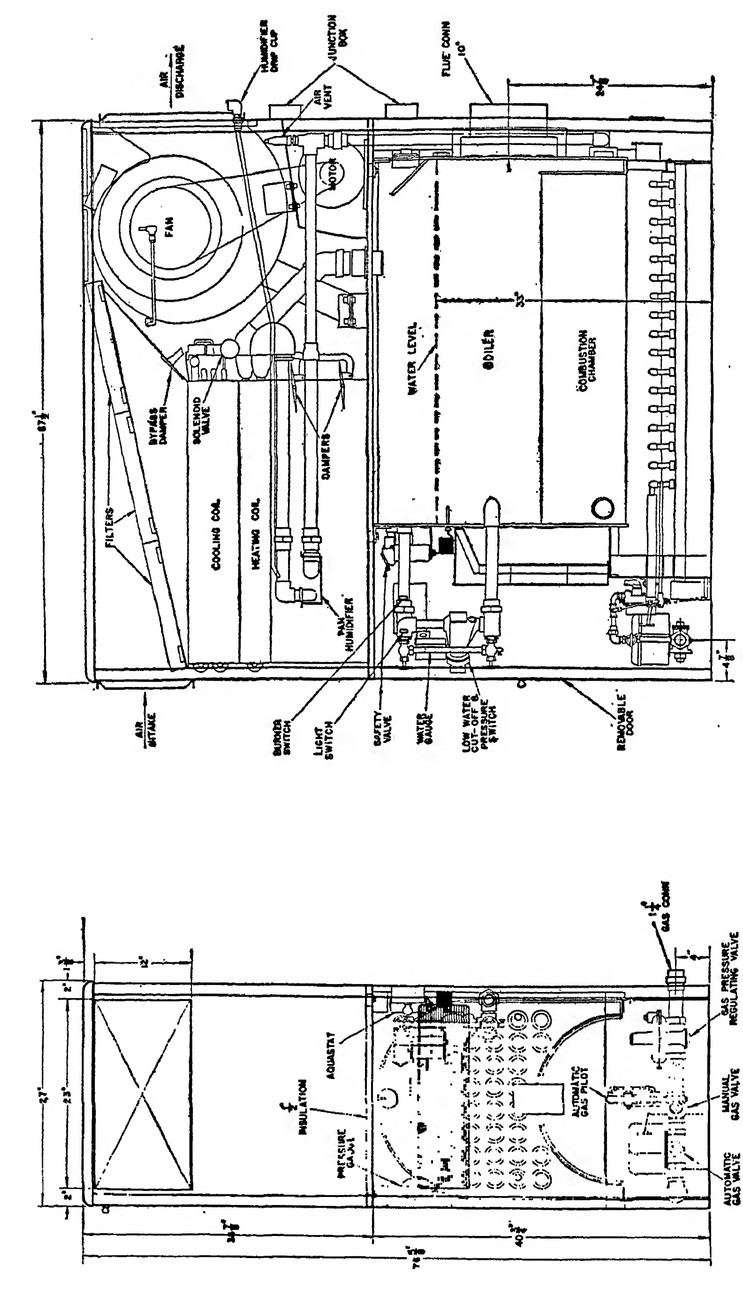


Fig. 177. Typical Gas-Fired Full Air-Conditioning Furnace

suming that air necessary for revitalization is provided by infiltration and leakage. The number of occupants in a residence of this size can well be assumed as 5 to 8. The incoming air by infiltration and leakage would be ample for that number of people. This incoming air offsets the vapor or humidity application to the air, therefore this fact must not be overlooked in the design calculations.

The type of apparatus is assumed as being similar to that shown in Fig. 177. Here steam is generated and used in a coil. The air is forced through this coil for heating. The humidification is provided by a special apparatus whereby an injector causes steam vapor to mix intimately with the air going to the occupied spaces.

Air is cooled by being forced through a cooling coil. The coil is connected to a refrigerating machine. The air is filtered both for summer and winter uses. In this example the sizes or amounts of filters, refrigeration machine, etc., will be calculated assuming that manufacturers' catalogues are to be used in the ultimate selection of apparatus. The steam boiler is assumed as operating at a maximum steam pressure of 5 pounds gauge. The entire system can be automatically controlled as explained in Chapter XII.

Humidification. The infiltration of air from the outside (temperature −10°F.) will require heating which will increase the volume and lower the relative humidity of that air. Also this incoming air in mixing with the recirculated air will decrease its relative humidity.

The amount of infiltration is small because of the weatherstrips and insulated surfaces. To determine the total heat transfer from the heating equipment, it is necessary to calculate the effects of the infiltration, which, as explained, lowers the relative humidity of the interior air unless some means is applied to artificially increase it. The following formula and calculations are used to determine the cubic feet of air from the outside to inside spaces and the quantity of vapor that is necessary for mixture with the air.

Refer to Table 92 and determine the lineal feet of cracks for the windows and doors. This total crackage is approximately 320 lineal feet. Formula **(59)** is now used to determine cubic feet of air entering due to infiltration.

$$C = \frac{H}{.018L(t_i - t_o)} \qquad \textbf{(59)}$$

where C = cubic feet of air per lineal foot of crack

H = the B.t.u.'s or the heat units per hour that are required for infiltration air.

.018 = the B.t.u.'s or the heat units for raising one cubic foot of air one degree F.

L = the total length of the crack opening of windows and doors.

t_i = the temperature of the interior

t_o = the temperature of the exterior

Referring to Table 92 the heat losses for the infiltration of air is determined. This is 10,772 B.t.u. per hour. Substituting in Formula (59)

$$C = \frac{10{,}772}{.018 \times 320 \times 80}$$

$$^*C = 23.4 \text{ cubic feet}$$

*No *leakage* is considered because with the insulation and new plaster the residence will be tight.

The 23.4 cubic feet multiplied by 320 feet equals 7,488 cubic feet of air per hour due to infiltration.

The 7,488 cubic feet of air is at a temperature of $-10°$F. The absolute humidity of air at this temperature with saturation conditions is .28 grains per cubic foot of air, assuming the relative humidity is 50 per cent. The grains of vapor is 50 per cent times .28 or .14 grains of vapor per cubic foot of air from the outside. The total grains of vapor is determined as follows. The 7,488 multiplied by .14 equals 1,048.32 grains.

The interior of the residence is maintained at 45 per cent relative humidity and 70°F. temperature. The absolute humidity is approximately 3.6 grains per cubic foot of air. To determine the total vapor for mixture with the infiltration to maintain 45 per cent relative humidity, it is necessary to find the difference of 3.6 and .14 grains which equals 3.46 grains.

The total grains of vapor that is added to the infiltration air is determined as follows. The 7,488 multiplied by 3.46 equals 25,908.48 grains of vapor. Then 25,908.48 divided by 7,000 grains per pound equals 3.701 pounds of vapor per hour to maintain 45 per cent relative humidity in the residence.

To make up or replace the water which is necessary due to the steam used for humidification, the supply is usually taken from the domestic hot water tank so as to avoid putting a strain on the boiler, which would happen if cold water was continually entering it. This water is usually about 140°F. To determine the B.t.u.'s or heat units that are necessary for vapor application, we will assume an average steam pressure of 2 pounds. Steam at this pressure has 1,153 B.t.u.'s per pound. Then subtracting $(140-32)$* from 1,153 equals 1,045 B.t.u.'s per pound of steam for vapor application. To determine the total B.t.u.'s for vapor application, the 1,045 B.t.u.'s per pound of steam is multiplied by 3.701 pounds of steam that is necessary for vapor additions to the air. This equals 3,867.545 B.t.u.'s per hour.

Radiator Surface. The square feet of area of the radiation surface needed to heat the air for distribution throughout the residence is determined by the use of Formula (60).

$$S=\frac{H}{E} \tag{60}$$

where S = the square foot area of the radiator
H = the total B.t.u.'s for heating
E = the B.t.u.'s that are transferred per square foot of radiator surface to the air which is 1,300 B.t.u.'s per square foot

The total heat loss 119,793 is taken from Table 92. Then 119,793 divided by 1,300 equals 92.15 square feet of radiation.

The steam generation that is necessary is determined by Formula (61).

$$W^s=\frac{H}{1000}+W^H \tag{61}$$

where W^s = the weight of steam per hour required for heating and humidification
H = the loss in B.t.u.'s
1000 = the B.t.u.'s per pound of steam that is transferred to the air
W^H = the weight of steam expressed in pounds per hour that are required for humidification

*Heat of liquid of feed water.

The 119,793 B.t.u.'s divided by 1000 equals 119.79 pounds of steam for heating per hour. This 119.79 plus 3.701 equals 123.491 pounds, the amount of steam that must be generated per hour by the boiler.

Gas Consumption. We will assume there are 800 B.t.u.'s per cubic foot of gas. Then the total B.t.u.'s needed per hour are 119,793 + 3,867.545 = 123,660.545. The following formula is used.

$$C.F.H.^G = \frac{H^T}{800E} \tag{62}$$

where $C.F.H.^G$ = cubic feet of gas per hour
H^T = total B.t.u.
800 = B.t.u.'s per cubic foot of gas
E = over-all efficiency of steam boiler—assume 75 per cent

Substituting in and solving Formula (62) gives 206.1 cubic feet of gas per hour which is the fuel consumption.

Volume of Air for Heat and Vapor Distribution. The volume of air necessary for heat and vapor distribution is determined by Formula (63).

$$Q^T = \frac{H^T}{.018(t_L - t_E)} \tag{63}$$

where Q^T = cubic feet of air per hour required
H^T = total B.t.u. for heating
.018 = B.t.u. to raise 1 cubic foot of air 1°F.
t_L = temperature of air leaving as 150°F.
t_E = temperature of air entering as 70°F.

The 119,793 divided by .018 (80) equals 83,419 cubic feet of air per hour. Then 83,419 divided by 60 equals 1,390.3 cubic feet of air per minute.

Fan. The fan necessary for the distribution of the air will be 1,390.3 cubic feet of air per minute capacity or the nearest commercial capacity listed in manufacturers' catalogues.

Duct Sizes. The duct sizes may be determined by using the Friction Chart, Fig. 70. To avoid considerable calculations, the chart may be used to determine the duct sizes and air velocities instead of the method given in the Technical Code. A single trunk duct system is used, see Fig. 173, together with branches as necessary. The friction of the air distribution system is .06 inch per 100 feet of length of duct. Each duct is designed separately to assure proper functioning of the system. In this connection the unit pressure drop method is used. This method avoids complicated measurements and provides accurate sizes according to air volume, air velocity, friction, and resistance.

The free area of the air intake from the outside of the filter, radiator, air inlet, and return and exit air openings are calculated according to the air volume and velocity for each section of the air-conditioning system. The air velocity through the air intake, from the outside is assumed to be 800 to 1000 feet per minute. The air velocity through the free area of the air filters is usually low to allow proper filtration. This velocity is 200 to 500 feet per minute. The filters for this example are assumed as sectional in design having facilities for cleaning. The air return opening and ducts are designed for full air capacity. The heating system is provided with an air duct of an area for full air capacity flow from the outside to the air intake opening of the equipment with control dampers installed for air flow control. Also a duct for the exit air flow from the

inside to the outside spaces is provided. This duct also has a damper. The outside air intake and the exit air ducts are necessary to provide facilities for the use of the heating equipment for ventilating with the air duct from the outside to the inside spaces and from the inside to the outside spaces.

The free area of the equipment and the sizes of the ducts are determined by use of the following formulas.

$$A = \frac{V^M}{V} \tag{64}$$

$$D^c \sqrt{\frac{A}{.7854}} \tag{65}$$

$$S = \sqrt{A} \tag{66}$$

$$W^R = \frac{A}{D^R} \tag{67}$$

$$D^R = \frac{A}{W^R} \tag{68}$$

where
A = area of ducts expressed as square inches or feet
V^M = the cubic feet of air per minute passing through the ducts
V = the air velocity expressed in feet per minute
D^c = the diameter of circular ducts expressed in inches or feet
S = the dimensions of a square form of air duct expressed in inches or feet
W^R = the width of the rectangular duct for the air passage
D^R = the depth of the rectangular duct for air passage

Air Intake. To determine the area of the air intake from the outside to the equipment, the 1,390.3 cubic feet of air per minute is divided by 800 (velocity) which equals 1.738 square feet. The free area of the radiation surface is the same as the air intake duct. Formula (64) was used.

Air Filters. To determine the free area of the filters, the 1,390.3 is divided by 200 which equals 6.95 square feet. Formula (64) was used.

Air Return and Exit Ducts. The air return and exit air ducts will have 1,390.3 cubic feet of air per minute passing from the inside spaces to the equipment and the outside spaces. The location, see Fig. 174, of the air return and exit air opening is in the side wall at floor level in the hall of the first floor. To determine the area of the air inlet opening of the air return and the exit air duct, the 1,390.3 is divided by 300 (air velocity through the inlet opening) which equals 4.634 square feet. The size of the inlet opening is calculated by using Formulas (65), (66), (67), and (68). The calculation for the rectangular ducts is accomplished by using Conversion Tables 57 and 58. One dimension is assumed, depending on the space allowed in construction details.

The size of the circular duct for the air return and exit from the inside spaces to the intake of the equipment is determined by the use of the Friction Chart, Fig. 70. The 1,390.3 cubic feet of air is located at the lower edge of the chart. This line is followed upward to the intercepting point with .06 inch of water horizontal line. From this intercepting point, follow the diagonal line for the duct size which shows a size of 18½ inches diameter. From the intercepting point of the cubic feet of air per minute vertical line and the .06 inch of water

friction horizontal line, trace the diagonal line for determining the air velocity per minute. This is 775 feet.

Note: If individual returns for each room are to be designed the procedure is identical.

Trunk Duct. This duct is designed exactly as for the return duct. Its size will be the same and have the same air velocity because the same amount of air passes through it.

Branch Ducts. To determine the sizes of the air supply branch ducts from the trunk duct to the various rooms, the cubic feet of air per minute that is required for each room is necessary. This is according to the B.t.u.'s required for each room. Formula (63) and the chart, Fig. 70, are used.

Referring to Table 92, it is seen that the heat loss for the living room is 8,906 B.t.u.'s per hour. This 8,906 B.t.u.'s is divided by 60 which equals 149.93 B.t.u.'s per minute required for the living room. To determine the cubic feet of air per minute necessary to heat the living room, Formula (63) is used. Substituting H^M in place of H^H

$$Q = \frac{H^M}{.018(t_L - t_E)}$$

The .018 is multiplied by the difference of air temperatures, 80°F., equals 1.44. Then dividing 149.93 by 1.44 equals 104.1 or the cubic feet of air per minute required for the heating of the living room. The size of the circular air duct for the living room is determined from the Friction Chart, Fig. 70, in the same manner as was explained for the return air duct. The duct is 6.8 inches in diameter. The air velocity is 385 feet per minute.

In like manner all other ducts for the various other rooms are determined. This information is shown in Tables 93 and 94.

To determine the size of the air inlet that is connected to the air supply duct, the following method is used. The 104.1 cubic feet of air vertical line is located at the lower side of the chart, Fig. 70. This vertical line is traced upward to the intercepting point with the 300 feet per minute diagonal line. From the intercepting point of the 104.1 and the 300 lines, trace the diagonal line representing the size 7.6 inches. This figure is found in Table 93. All other inlets are determined in like manner.

When selecting grilles for inlet openings, the over-all dimensions will have to be increased so as not to restrict the free opening. Grilles can be selected having the necessary free area. See Chapter VIII.

All circular ducts can be converted to rectangular ducts by referring to the conversion tables.

The bathroom, for example, requires so little air per minute for heating that complications arise in using the Friction Chart, Fig. 70. The chart, as shown, is limited to 40 cubic feet per minute. However, the lines of the chart may be extended and then be used for the bathroom, hall, etc. The size of the air duct and air inlet openings and the velocity for the bathroom and second floor hall can be calculated by using Formulas (64), (65), and (66).

Stacks. The Technical Code, Article 6, Section 2, Item j gives a method for sizing wall stacks. An investigation relative to the conditions in this example would show that 12″×3½″ stacks would be satisfactory.

Location of Grilles. The hot air grilles (registers) should be located about 6′8″ above the floor for all first floor rooms and 6′4″ for all second floor

Table 93. Design Data for Ducts

Room	Cubic Feet per Minute	Inches of Water Pressure Drop per 100 Feet of Duct	Sizes of Ducts in Inches (Diameter)	Sizes of Openings Air Inlets Inches	Velocity of the Air Feet per Minute
Living Room	104.1	.06	6.8	7.6	385
Hall First Floor	82.80	.06	6⅜	6.8	350
Dining Room	144.00	.06	7.9	8.9	420
Sun Parlor	302.00	.06	10¼	13.0	520
Kitchen	111.80	.06	6.9	7.8	390
Bedroom No. 1	66.68	.06	5.8	6⅛	330
Bedroom No. 2	88.30	.06	6.5	7	360
Bedroom No. 3	92.20	.06	6.7	7⅛	365
Bedroom No. 4	66.10	.06	5.8	6⅛	330
Hall Second Floor	21.03	.06	3.24	3.5	350
Bathroom	21.60	.06	3.288	3.54	350
Playroom	288.60	.06	10⅛	12¾	514

Table 94. Miscellaneous Data

Sections of the Air Passages System	Cubic Feet of Air per Minute	Inches of Water Pressure Drop per 100 Feet of Duct	Size of Duct in Inches	Air Velocity in Feet per Minute	Size of Openings for Air Inlets Inches
Return Duct	1,390	.06	18½	775	29
Exit Duct	1,390	.06	18½	775	29
Outside Air Inlet Opening	1,390	.06	18½	775	29
Section A* Trunk Duct	291.7	.06	10⅛	514	..
Section B* Trunk Duct	523.48	.06	12½	595	..
Section C* Trunk Duct	799.68	.06	14¾	670	..
Section D* Trunk Duct	1,390	.06	18½	775	..
Main Air Duct	1,390	.06	18½	775	29

*See Fig. 173.

rooms. The cold air return grille should be in the wall as close to the floor level as possible.

Duct Layout. Fig. 173 shows the duct layout. In the example of Chapter VI the investigation and study relative to the layout of another typical layout was explained. Therefore at this point the only added explanation needed is that the main trunk should be so located as to make all branches as short and free from turns as possible.

Volume Dampers. There are many factors of climatic and occupancy conditions that affect the correct division and supply of air to the various rooms. To better control the operation of the whole system, volume dampers for each reduction of the main branch duct are supplied. Such dampers are set following

a complete test of the system. This allows the required control of the air for each room and makes it possible to change the supply conditions as required.

Cooling and Dehumidifying. To determine the heat and vapor gains for this residence (Figs. 173 to 176), it is necessary to consider many factors. This data must be calculated and put into such a form as to be easily used by the designer of the equipment. The following data or requirements are typical.

GENERAL DATA

Purpose of Building		Residence
Exposed Direction		South
Type of Air-Conditioning System		Central
Outside Temperature		95°F.
Relative Humidity		50 per cent
Inside Temperature		80°F.
Relative Humidity		50 per cent
Temperature on Outside Due to the Sun	on Side Walls	115°F.
	on Roof	150°F.
Temperature of Air Entering Residence	Heating Season	−10°F.
	Cooling Season	95°F.
Desired Relative Humidity	Heating Season	45 per cent
	Cooling Season	50 per cent
Number of Occupants		8
Cubic Feet of Air per Minute	Heating	1390
	Cooling	1674 to 2050
Wind Velocity and Direction	Heating Season	S.W.—15 M.P.H.
	Cooling Season	N.E.—15 M.P.H.
Types of Windows		Double Hung Casement
Infiltration		23.6

Table 95. *U* Values

Surface	Material	Thickness Inches	Insulation Insulite Thickness in Inches	*U* Value	
				Heating	Cooling
Glass	Wood Shingles Awnings	..	..	1.13	1.13
Walls	Wood Insulite	6	½	.19	.19
Roofs	2x8 Rafters Insulation Shingles	..	⅞	.10	.10
Top Floor Ceiling	2x8 Rafters Insulation	..	½	.21	.21
Basement Floor	Concrete on Cinders	4	..	1.07	1.07
Basement Ceiling	No Plaster	..	..	.34	.34

Table 96 shows the calculation sheet for calculating heat gains. The areas of walls, glass, ceiling, doors, etc., the window and door cracks, and heat transfer per square foot per hour are all shown in details and ready for instant use.

***Table 96. Calculation Sheet Showing Basic Figures Used in Estimating Heat Gains for the House Shown in Figs. 173 to 176 Inclusive**

Rooms of Residence	Net Exposed or Other Area Sq. Ft.	U Values	Temp. Diff. °F.	Lineal Feet of Crack	Loss in B.t.u. per Hour	Totals
Living Room						
Wall	264	.19	15 and 35†	..	1,247	
Floor	...	....	..	..	.....	
Ceiling	...	....	..	..	.....	
Glass	41	1.13	15 and 21†	..	844	
Door Crack	...	....	..	..	.....	
Window Crack	...	....	15	35	223	
Door	...	....	..	..	.....	2,314
Hall 1st Floor						
Wall	101	.19	35	..	672	
Floor	78	.34	7.5	..	199	
Ceiling	...	....	..	..	.....	
Glass	...	....	..	..	.....	
Door	43	1.13	35	..	1,700	
Door Crack	...	....	15	25	159	
Window Crack	...	....	..	..	.....	2,730
Dining Room						
Wall	203	.19	15 and 35	..	1,016	
Floor	178	.34	7.5	..	454	
Ceiling	...	....	..	..	.....	
Glass	63	1.13	15 and 21	..	1,198	
Door	...	....	..	..	.....	
Door Crack	...	....	..	..	.....	
Window Crack	...	....	15	49	312	2,980
Sun Parlor						
Wall	222	.19	15 and 35	..	1,271	
Floor	136	.34	15	..	693	
Ceiling	136	.62	70	..	5,903	
Glass	108	1.13	15 and 21	..	2,360	
Door	...	....	..	..	.....	
Door Crack	...	....	..	..	.....	
Window Crack	...	....	15	69	450	10,677
Kitchen						
Wall	199	.19	15 and 35	..	1000	
Floor	129	.34	7.5	..	329	
Ceiling	...	....	..	..	.....	
Glass	20	1.13	15 and 21	..	408	
Door	12	1.13	35	..	281	
Door Crack	...	....	15	18	115.5	
Window Crack	...	....	15	18	115.5	2,249
Bedroom No. 1						
Wall	210	.19	35	..	1,397	
Floor	...	....	..	..	.....	
Ceiling	168	.21	28	..	988	
Glass	24	1.13	15 and 21	..	559	
Door	...	....	..	..	.....	
Door Crack	...	....	..	..	.....	
Window Crack	...	....	15	17	110	3,054
Bedroom No. 2						
Wall	229	.19	35	..	1,523	
Floor	...	....	..	..	.....	
Ceiling	190	.21	28	..	1,120	
Glass	25	1.13	15 and 21	..	582	
Door	...	....	..	..	.....	
Door Crack	...	....	..	..	.....	
Window Crack	...	....	15	19	120	3,345
Bedroom No. 3						
Wall	203	.19	15 and 35	..	1,021	
Floor	...	....	..	..	.....	
Ceiling	159	.21	28	..	935	
Glass	26	1.13	15 and 21	..	530	
Door Crack	...	....	..	..	.....	
Window Crack	...	....	15	19	120	2,606

***Table 96—Continued**

Rooms of Residence	Net Exposed or Other Area Sq. Ft.	*U* Values	Temp. Diff. °F.	Lineal Feet of Crack	Loss in B.t.u. per Hour	Totals
Bedroom No. 4						
Wall	157	.19	..	..	721	
Floor	...	....	..	..	.....	
Ceiling	86	.21	..	..	504	
Glass	18	1.13	..	..	349	
Door	...	....	..	..	.....	
Door Crack	...	....	..	..	.....	
Window Crack	...	....	15	16	102	1,676
Bathroom						
Wall	45	.19	35	..	298	
Floor	...	....	..	..	.....	
Ceiling	49	.21	28	..	280	
Glass	4	1.13	15 and 21	..	93	
Door	...	....	..	..	.....	
Door Crack	...	....	..	..	.....	
Window Crack	...	....	15	8	34	705
Hall 2nd Floor						
Wall	...	....	..	..	.....	
Floor	...	....	..	..	.....	
Ceiling	103	.21	28	..	616	
Glass	...	....	..	..	.....	
Door	...	....	..	..	.....	
Door Crack	...	....	..	..	.....	
Window Crack	...	....	..	..	.....	616
Playroom						
Wall	335	.19	10	..	1,140	
Floor	297	1.07	5	..	1,590	
Ceiling	...	....	..	..	.....	
Glass	5	1.13	15	..	89	
Door	16	1.13	15	..	281	
Door Crack	...	....	15	9	54	
Window Crack	...	....	15	18	118	
Partition	150	.62	7.5	..	698	3,970

B.t.u's Total	36,932
Lighting	5,964
Range Kitchen	1,200
Occupants	3,200
B.t.u.'s Total	47,296
Sun Effect on One Side of Building	2,095
Net Heat Gains Sensible Heat B.t.u.'s Total	45,201
Latent Heat	4,601
B.t.u.'s Sensible and Latent Heat Total	49,802

The living room has an exposed wall surface of approximately 264 square feet. The *U* value is .19. The temperature difference between the inside and the outside will be according to the exposure to the sun and the temperature of the outside air. Therefore the heat gains are affected or controlled according to the sun effects. By studying the drawings, it can be seen that the living room has a S.E. and S.W. exposure to the sun. According to approximate scaling the S.E. and S.W. exposures are 2′6″ at front sections on the east side of the sun parlor and 12′9″ for the east or west wall section. The S.E. exposure is 12′9″ +2′6″ or 15′3″. Then 15′3″ multiplied by 10′0″ (ceiling height) equals 152.5 square feet area. The S.W. exposure is also 152.5 square feet. Therefore it is necessary to determine which side or area will probably receive the greatest sun effect. In this case both areas exposed to the sun are equal in area. The

*The figures shown in this table are *approximate* in many cases, following actual practice. Most engineers feel that the small error caused by using approximate figures is more than made up for in other ways.

†Temperature difference for exposure to outside air and to sun effect.

window area must be subtracted from the wall area. The wall area is (for each side) 152.5 square feet. The two areas combined equal 305 square feet. The glass area is 41 square feet so the net wall area is 305−41=264 square feet, as shown in Table 96. We must consider two different temperature differences. One for the area exposed to the sun and one for the area exposed to the outside temperature. From General Data it is seen that the sun maintains a temperature of 115°F. on the wall exposed to it. The inside temperature is maintained at 80°F. Then 115°−80°=35°F. This is shown in Table 96. The temperature difference for the wall exposed to the outside temperature is 95°−80°=15°F. This is also shown in Table 96.

The actual heat transmission or gain can be calculated by using Formula (14), Vol. II.

$$H_t = AU(t_o - t)$$

Substituting $H_t = 130^* \times .19 \times 35^\circ$

$$H_t = 864 \text{ B.t.u. per hour}$$

This is the heat gain due to the sun effect.

Substituting again in Formula (14), Vol. II.

$$H_t = 134^* \times .19 \times 15^\circ$$

$$H_t = 383 \text{ B.t.u. per hour}$$

This is the heat gain due to the outside temperature.

Then 878+376=1,247 B.t.u. total heat gain through walls of living room.

The heat gain through the windows due to the surface temperature difference and the infiltration of air from the outside are calculated next. From the plans it can be determined that the windows on the east side of the living room total 22 square feet and that the window on the west side contains 19 square feet. The east windows are the largest area so they will be assumed as being affected by direct sun rays. The temperature difference would be assumed as 115°−80°=35°F. However, the windows have awnings† over them and these reduce the temperature caused by the sun.

With the awnings in place the temperature caused by the sun on the glass would be reduced. Here we will assume that the temperature difference for the windows exposed to the sun is 21°F.

Note: Opinion varies as to the procedure of determining sun effect on windows. Vol. II, Chapter VI gives one method which may be used. Some engineers favor other empirical methods. In this example it is assumed that the awnings reduce the temperature difference to 60 per cent of 35°. Thus, 35°×.60=21°F.

The heat gain is calculated as for the walls by using Formula (14), Vol. II.

$$H_t = AU(t_o - t)$$

Substituting $H_t = 22 \times 1.13 \times 21$

$H_t = 522$ B.t.u., heat gain per hour through windows exposed to the sun

The heat gain for the west window is, in like manner, 19×1.13×15=322 B.t.u. per hour. Then 522+322=844 B.t.u., total heat gain through the windows.

*Difference in 130 and 134 due to difference in glass area on two sides.

†It should be kept in mind that all windows may have awnings, except on the north side. However, for the living room, as an example, the sun does not shine on both east and west windows at the same time so only one side is considered for sun effect. Also it should be remembered that while the sun shines on the east windows the west windows are exposed to the 95°F. air temperature. In calculating heat gains we must always remember to calculate the *maximum* that may be encountered so that the cooling apparatus will have sufficient capacity.

The air coming in by infiltration through the cracks around the windows will result also in a heat gain. The temperature difference is 15°F. Only the crack on one side of the living room is used which is 35 feet. We can substitute in Formula (37) as follows.

$$\text{Heat gain} = .018CL(t_i - t_o)$$
$$= .018 \times 23.6 \times 35 \times 15$$
$$= 223 \text{ B.t.u., heat gain due to infiltration}$$

The total heat gain for the living room is therefore,

$$1{,}247 + 844 + 223 = 2{,}314 \text{ B.t.u.}$$

Note: The playroom under the living room, the bedrooms over, and the first floor rooms on either side are all cooled to 80°F., and so no further heat gains occur than as shown just above.

The calculations to determine the heat gains for the other rooms of the residence are similar to the methods used for the living room. The following factors should be applied for heat gain for the various other rooms of the residence.

The sun parlor has a S.E. and S.W. exposure to sun effects and the air temperature of the outside. The S.E. and the S.W. exposures are of equal area and either exposure can be used for the calculation of heat gain due to sun effects and the same method can be applied for the calculation of the heat gain due to effects from the air temperature as shown for living room.

The sun parlor has a roof which is exposed to the direct sun effects. This roof surface will have a temperature of 150°F. There is very little attic space between the roof and the ceiling. The floor of the sun parlor is exposed to the air temperature of the outside.

There are three exposures of the windows in the sun parlor. If the plans of Fig. 173 are checked, the front window will be found to have the largest area and should be considered for the infiltration losses. The kitchen and the dining rooms each have two window spaces in the walls and the window of the largest area should be used for the calculation of the infiltration gains. The walls and the windows of the north side of the kitchen and the dining rooms are not affected by the sun.

The exposed wall surfaces of the playroom that are above the ground level include the south, east, and west walls. One inside wall or partition is exposed to spaces that are without air conditioning. A temperature is assumed such as is listed in the Code, or 86.75°F.

The parts of the wall and floor surfaces that are in contact with the earth will have a temperature of 5°F. more than the normal inside temperature or 85°F. The basement walls below and above the grade level will have different heat gains due to the temperature variations for each section. The wall surfaces below grade have a 10°F. difference of temperature from the inside of the residence, or 90°F. The playroom walls have a small area above the grade and these wall surfaces are calculated for heat transfer on this basis.

The calculations for the heat gain of the rooms of the second floor of the residence should include sun effects of the attic spaces and the roof surfaces.

Careful analysis of the plans and the elevations of the residence should be maintained to determine the actual area of the roof surface that is affected by the sun and the outside air temperature, as the calculations for heat gain are based on these temperatures. The two front bedrooms are a S.E. and S.W. exposure and the sun will affect both wall and window of these rooms at the same

period. To determine the maximum heat gain for the two front bedrooms, sun effects on both walls and windows of each room are considered.

To determine the infiltration losses for the two front bedrooms, the window of each room that has the largest crackage is used for the calculation.

The two rear bedrooms are N.E. and N.W. exposures. The sun will not affect the wall and window surfaces on the north side of the bedrooms and the heat gain calculations are according to the temperature of the outside air. The sun will affect the east and west walls and windows during a part of the day so the temperature condition, due to this sun exposure, is considered for the heat gain for these bedrooms.

To determine the infiltration losses for the two rear bedrooms, the window of each bedroom that has the largest crackage is used for the calculations.

The inside of the residence has 100 per cent recirculation and the air that is necessary for the ventilation is of a small quantity. The number of occupants is low and it is practical to assume that the air infiltration through windows, doors, and by the frequent entrances and exits of the occupants is ample.

For the first floor hall the temperature difference for the floor is 7.5° F. The hall temperature at the floor level is 80°F. This temperature (that part of basement without cooling) is determined by adding the outside and inside temperatures and dividing by 2. Thus 95°+80°=175° and 175° divided by 2 is 87.6°.

To determine the heat gain for the main entrance door, we assume that the door is exposed to the sun and therefore has a 115° temperature on its outside surface. Formula (14), Vol. II, is used and the heat gain is $1.13 \times 43 \times 35 =$ 1,700 B.t.u. per hour. The door infiltration is calculated as for a window.

To determine the heat gain due to the second floor ceiling, it is first necessary to calculate the attic temperature. This can be done by using Formula (69).

$$t_a = \frac{t_e U_c + N t_o U_r}{U_c + N U_r} \tag{69}$$

where t_a = temperature of attic

t_e = temperature at inside surface of ceiling as 80°F. in this example

U_c = coefficient of heat gain per square foot of the ceiling per degree difference of temperature, or in this example .21

N = the ratio of roof area to ceiling area, or in this example 1.5

U_r = coefficient of heat gain for roof and ceiling surface, or in this example about .10 for such a pitch

t_o = temperature of exterior of roof surface, or 150°F.

Substituting $t_a = \dfrac{80 + .21 + 1.5 \times 150 \times .10}{.21 + 1.5 \times .10}$

$t_a = 108$°F.

This temperature of 108°F. is the attic temperature and the temperature difference for the second floor ceiling would be 108°−80°=28°F.

The latent heat transfer or gain due to vapor from various sources of the inside of the residence cannot be calculated for each separate space. It is the usual practice to determine the latent heat transfer for the entire residence.

The heat transfer from the occupants to the inside air of the residence is calculated for the air conditioning of the entire residence. The occupants will not remain in one room for any length of time and the living room is usually

occupied at the maximum capacity more than the other rooms. The sensible heat gain and the latent heat gain from the occupants and other sources are added to the total heat gain for living room. This will provide ample heat and vapor transfer with the air distribution from the equipment to the inside spaces of the residence to maintain comfortable and healthful occupancy conditions.

There are 8 occupants of the residence. The average heat gain from each occupant, with the usual activities maintained, is 400 B.t.u.'s per hour per occupant. The total heat transfer from the occupants is 400×8 which equals 3,200 B.t.u.'s per hour.

The heat gain from the lighting system to the inside air is about 3.4 B.t.u.'s per watt rating of the lamps. The usual lamp ratings for the residence building is about 1 watt per square foot of floor area. The residence has about 1,750 square feet of floor area. The heat transfer due to the lighting system with maximum illumination is $1{,}750 \times 1 \times 3.4$ equals 5,963.6 B.t.u.'s, the heat gain to the air from the lighting system.

The other heat and vapor gains are very small and it is the usual practice to eliminate them or provide an average B.t.u. requirement according to the method of design of the air-conditioning system.

The air as a medium for heat transfer passing over and around the surfaces of the equipment will require cooling for the transfer of the heat quantity necessary to maintain the inside of the residence at 80°F. and 50 per cent relative humidity. The sensible heat transfer capacity of the equipment is determined by the following formula and calculations.

$$\text{Tons of refrigeration or equivalent cooling} = \frac{H}{12{,}000} \tag{70}$$

$$\frac{45{,}201}{12{,}000} = 3.77 \text{ tons of refrigeration or equivalent cooling}$$

12,000 equals the B.t.u.'s transfer per hour that is equivalent to one ton of ice or refrigeration capacity.

The heat gain of 45,201 B.t.u.'s per hour for the inside of the residence will require air distribution at a temperature that will be adaptable for the comfortable and healthful occupancy conditions.

The factors that are effective for the proper quantity of air for air conditioning are the temperature difference of the air to and from the equipment and the inside of the residence and the heat gain as the B.t.u.'s per hour.

To determine the cubic feet of air per minute that is required for cooling or heat removal from the inside of the residence the following formula is used.

$$\text{C.F.M.} = \frac{H}{.018(t_E - t_L)60}$$

where H = net heat gain
t_E = air temperature entering equipment
t_L = air temperature leaving equipment

Substituting $$\text{C.F.M.} = \frac{45{,}201}{.018(80-60)60}$$

$$\text{C.F.M.} = 2{,}050$$

This volume of air is larger than required for heating and as the fan was originally selected according to the C.F.M. required for heating it can be seen

that the fan will have to be speeded up to provide 2,050 C.F.M. during the cooling season. A practical solution of this condition is to increase the temperature difference as (80° − 55°) or a 25°F. difference of the air entering and leaving the inside of the residence. With such a difference between the air entering and leaving the equipment, it is necessary to increase the capacity of the equipment used for heating and humidification if the same equipment is to be used for cooling and dehumidification. The increase of air volume is provided, as specified, by speeding up the fan. This can be shown by the following formula.

$$\text{C.F.M.} = \frac{H}{.018(t_E - t_L)60}$$

$$= \frac{45{,}201}{.018(80-55)60} = 1{,}674$$

This is cubic feet of air per minute for cooling and dehumidification.

The coils or surfaces for heat transfer will require a capacity for transfer of sensible heat of 45,201 B.t.u.'s per hour and a free area of 1,674 or 2,049 cubic feet according to the temperature difference that is maintained. Such a coil can be selected from catalogues published by manufacturers, or by such data as shown in Figs. 93 and 94.

The air infiltration from the outside to the inside spaces of the residence result in an increase of the vapor quantity of the air.

This additional vapor quantity will require a heat transfer from the air for the removal of the vapor that is in excess or that is gained by the air passing from the outside at 95°F. and 50 per cent relative humidity to the inside and mixing with the air that is maintained at 80°F. and 50 per cent relative humidity. The occupants of the residence will eliminate about 700 grains of moisture vapor per hour per occupant with the conditions as maintained for the residence. The heat quantity that is required for the vapor increase from the outside and from the occupants is the latent heat requirements for the air-conditioning application.

There are 8 occupants of the residence and each occupant will add 700 grains of vapor per hour to the inside air. The 700 multiplied by 8 equals 5,600 grains of vapor per hour added to the inside air from the occupants.

There are 7,488 cubic feet of air per hour passing from the outside to the inside of the residence. The air quantity is at 95°F. and 50 per cent relative humidity and is mixing with the air of the inside that is maintained at 80°F. and 50 per cent relative humidity.

The mixture of the air from the outside with the air of the inside will result in an increase of vapor quantity per cubic foot. Therefore to maintain comfortable conditions, it is necessary to remove the excess vapor. The elimination of the excess vapor is accomplished by the heat transfer from and to the air volume and the equipment of the system.

The air from the outside spaces at 90°F. and 50 per cent relative humidity will have 9 grains of vapor per cubic foot. The air of the inside has 5.8 grains of vapor per cubic foot with conditions of 80°F. and 50 per cent relative humidity. Therefore it is necessary to remove the difference of (9 − 5.8) grains of vapor per cubic foot of air which equals 3.2 grains of vapor.

This 7,488 cubic feet of air multiplied by 3.2 grains equals 23,961.6 grains per hour that is removed from the infiltration air. There are many other items

of servicing and occupancy that may add to the vapor quantity. From such miscellaneous origins about 3,000 grains per hour is assumed.

The total grains that must be eliminated from the air to maintain 80°F. temperature and 50 per cent relative humidity is determined by adding 5,600+23,961.6+3,000 grains which equals 32,561.6 grains per hour.

We will assume that it is necessary to have 2,000 cubic feet of air per minute or 120,000 cubic feet of air per hour passing to and from the inside of the residence and the air-conditioning system.

The vapor that is added per cubic foot of air (that is from the occupants, the infiltration air and other sources) is determined by dividing the 32,561.6

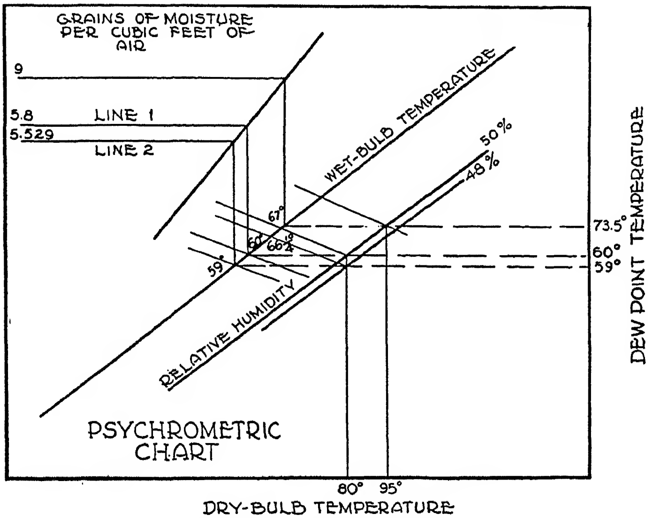


Fig. 178. Abbreviated Psychrometric Chart

grains by the 120,000 cubic feet of air per hour (the total air volume required for the air-conditioning application) which equals .271 grains of vapor that is added per cubic foot of air. Therefore to maintain the air of the inside spaces at 80°F. and 50 per cent relative humidity, it is necessary to remove .271 grains of vapor per cubic foot of air passing to and from the equipment and the inside of the residence.

To determine the condition of the air leaving the equipment to maintain the inside spaces at comfortable temperature and humidity, refer to the Psychrometric Chart, Fig. 178. If .271 grains must be removed, then line 1, Fig. 178, represents the moisture for air at 80°F. and 50 per cent relative humidity (5.8 grains) and line 2 represents 5.8−.271 or 5.529 grains. Then back tracing the 5.529 (line 2) it is seen that air entering the inside should be 48 per cent relative humidity, 66¼° wet bulb and 59° dew point. This indicates that the air temperature is lowered to 59°F. to remove 32,561.6 grains of vapor per hour from the 120,000 cubic feet of air per hour that is passing over and around the

air-conditioning equipment. Therefore the air temperature of 55°F. that is required for the transfer of sensible heat is ample for vapor removal.

The latent heat transfer that is required for the removal of the 32,561.6 grains of vapor from the air per house is determined as follows.

The 32,561.6 grains of vapor is divided by 7,000 grains (per pound) which equals 3.423 pounds of vapor per hour which is removed from the air passing to the equipment from the inside of the residence.

To remove one pound of vapor from the air with the average conditions maintained, it is necessary to have a transfer of 1,050 B.t.u.'s. The 3.423 pounds of vapor per hour multiplied by 1,050 B.t.u.'s equals 4,601 B.t.u.'s per hour which is the heat transfer from and to the air and the air-conditioning equipment for the excess vapor removal from the air of the inside of the residence.

The total heat transfer that is required from the air and the air-conditioning equipment is the sensible heat, which is 45,201 B.t.u.'s, plus the latent heat, which is 4,601 B.t.u.'s. This equals 49,802 B.t.u 's which is the total heat transfer per hour for maintaining the inside spaces at 80°F. with 50 per cent relative humidity, and with a 95°F. and 50 per cent relative humidity on the outside.

The coils of the air-conditioning equipment for the heat transfer from the air for the air-conditioning application will require a surface that is adequate for the transfer of 49,802 B.t.u.'s per hour. This is the total heat (sensible and latent) transfer requirements for the heat gains due to climatic and occupancy conditions and the various sources of vapor gains of the inside of the residence.

The maximum capacity of the air-conditioning equipment for cooling and dehumidification of the air for the inside of the residence is determined as follows. The 49,802 B.t.u.'s per hour (the total sensible and latent heat gains) is divided by 12,000 (the B.t.u.'s transfer equivalent of one ton of ice) which equals 4.15 tons of refrigeration or equivalent heat transfer capacity of the air-conditioning equipment.

Summary. The system described in the foregoing example provides for the complete or year around air conditioning. The necessary apparatus can be selected from manufacturers' catalogues. The stacks and most of the duct work is of standard or stock sizes which can be purchased ready to install. The system, because of the outside air inlet, can be employed in such a manner as to use cool outside night air for natural cooling whereby the cooling apparatus is not used. Such a procedure could be used during cool nights when the outside temperature drops to 70°F. or lower.

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